Westinghouse Design Report

Thermoelectric Environmental Control Unit

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WESTINGHOUSE ELECTRIC CORPORATION
ATOMIC EQUIPMENT DIVISION
CHESWICK, PENNSYLVANIA

THERMOELECTRIC ENVIRONMENTAL CONTROL UNIT

FINAL DESIGN REPORT

U.S. AERDL Contract No. DA-44-009-AMC-1136(T)

Task No. IM 643303D54503

October 8, 1965

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WESTINGHOUSE ELECTRIC CORPORATION
ATOMIC EQUIPMENT DIVISION
CHESWICK, PENNSYLVANIA

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NOTICE

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TABLE OF CONTENTS

1.0 LIST OF ILLUSTRATIONS

2.0 INTRODUCTION SUMMARY

This report is the final product of an Engineering Program performed by Westinghouse Electric Corporation for the design of a thermoelectric environmental control unit. The work was performed under contract number DA-44-009-AMC-1136(T) for the U.S. Army Engineer Research and Development Laboratories, Fort Belvoir, Virginia.

The unit capacity is 24,000 Btu/hr when operating in the cooling mode. The unit is self contained, transportable and contains power conversion and automatic thermostatic control for both cooling and heating.

The objective of the program was to produce an optimum practual unit by applying thorough, objective, and imaginative study procedures. The procedure used in arriving at the final design is illustrated by the Information Flow Diagram, Figure 2-1. The basic engineering tasks included analysis, parametric studies and final engineering design.

A reliable high performance design has been achieved through simplified design, integrated design and ingenious arrangement and packaging. The general arrangement of the major components and the air flow is shown by Figure 7-7.

The basic building block for the thermoelectric core is a nominal 1000 Btu/hr module.

The module is the Westinghouse "Direct Transfer" type which is the key to economical, reliable, high performance thermoelectric cooling systems. The modular design of the thermoelectric core permits easy servicing and interchangeability with all sizes of air conditioners.

Three duplicate fans are used for air circulation. The motors are dual wound to permit operation on 60 or 400 cycle power sources. Operation on 50 cycle power is also permissible at reduced speed and cooling capacity.

A novel power conversion package is housed in the assembly which will also accept input power at frequencies from 50 cycles up.

It incorporates automatic proportional temperature control between 0 and full power. Smooth dc current is delivered to the thermoelectric elements over the entire control range which results in high performance at all loads.

The results of the design studies and the final design are presented in the subsequent sections of this report.

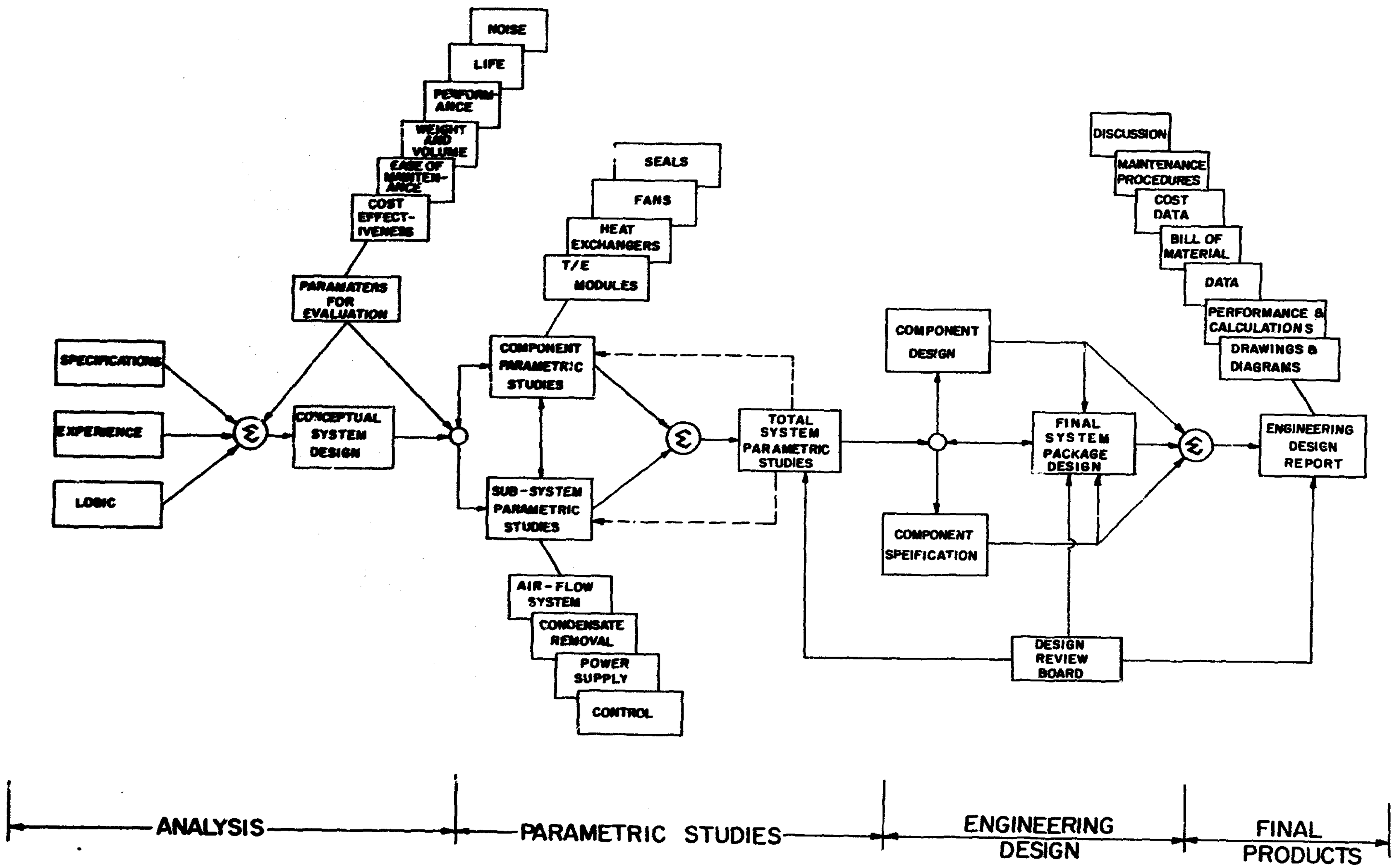


FIG. 2-1
INFORMATION FLOW DIAGRAM

CSM 1182

3.0 DESIGN STUDIES

3.1 Introduction

The environmental control unit design was approached on an integrated system basis in which the thermoelectric core, power supply, air moving equipment and packaging were considered as a system. The principal factors used as a basis for evaluating the design included the following:

1. Cost
2. Performance
3. Volume
4. Weight
5. Reliability
6. Maintainability
7. Noise
8. Versatility

In the design study phase, basic design concepts were evolved and parametric data for components and systems were generated. These were compared and evaluated to determine which most completely met the objectives of the study.

The following discussion develops the background and major steps which lead to the final design.

3.2 The Design Plan

As an initial step, a design plan was developed which served to direct the course of engineering study. The design plan set overall guide lines for system characteristics, component and sub-unit requirements to assure compatible operation between sub-systems and consistency in overall design.

The design plan included several candidate design concepts for which an analysis was carried to the point where the superior system became apparent. The major design concepts varied with respect to the basic approach to the thermoelectric module and the air flow system. The four concepts studied are as follows:

1. Conventional - this system employs a thermoelectric module in which insulation is interposed between the TE couples and the heat exchangers.

2. Direct Transfer - this system uses no insulation in the heat transfer path.
3. Counterflow - in this system the heat exchangers were arranged for vertical inside and outside air flow.
4. Crossflow - the heat exchanger and package configuration is arranged for horizontal outside air flow and vertical inside air flow.

In addition to the principal factors for system evaluation, such as cost, performance, volume and weight, the following were included as desirable objectives:

1. Module Standardization - Modules with ratings and power supply requirements to permit their use as a building block for all sizes of air conditioners.
2. Reliability - to withstand environmental stress including mechanical and thermal shock encountered in military service.
3. Component Interchangeability - minimum number of component types such as fan models and module types.
4. Versatility - to permit application on 60 cycle and 400 cycle power sources with minimum modification.

3.3 Performance Considerations

The overall performance of a thermoelectric cooling system is measured in terms of COP (coefficient of performance) which is the ratio of the heat pumped to the power input in equivalent units. The system power input includes the power to the thermoelectric elements, the power to drive the fans and the additional input power required to off-set the system losses. Performance optimization involves a relatively complex study of the variables which will produce rated cooling capacity with minimum total power.

3.3.1 Thermoelectric Couple Performance

The performance of a thermoelectric heat pump is basically a function of the temperature difference over which heat is transported.

The basic relationships which determine steadystate performance of a thermoelectric couple are as follows:

$$Q_c = \alpha I t_b - 1/2 I^2 \rho L/A - k A/L \; \Delta t_p \qquad (1)$$

$$P = I^2 \rho L/A + I \alpha \Delta t_p \qquad (2)$$

$$COP = \frac{\alpha I t_b - 1/2 I^2 \rho L/A - k A/L \; \Delta t_p}{I^2 \rho L/A + I \alpha \; \Delta t_p} \qquad (3)$$

where

Q_c = heat pumped, watts

P = power input, watts

COP = coefficient of performance

α = thermoelectric power, volts/°F

t_b = cold junction temperature, °F absolute

I = current, amperes

ρ = pellet resistivity, ohm-in.

L = pellet length, in.

A = pellet area, in.^{2}

k = pellet thermal conductivity, watts/in.-°F

t_p = temperature difference between pellet hot and cold junctions, °F

R = ρ L/A, electrical resistance of the pellet, ohms

K = k A/L, thermal conductance of the pellet, watts/°F

The term Δt_p is a function of the difference between the heat source and the heat sink and the heat transfer properties at the hot and cold junctions. This is illustrated by the thermal schematic and the temperature relationship of the thermoelectric device, Figure 3-1, and Figure 3-2 respectively.

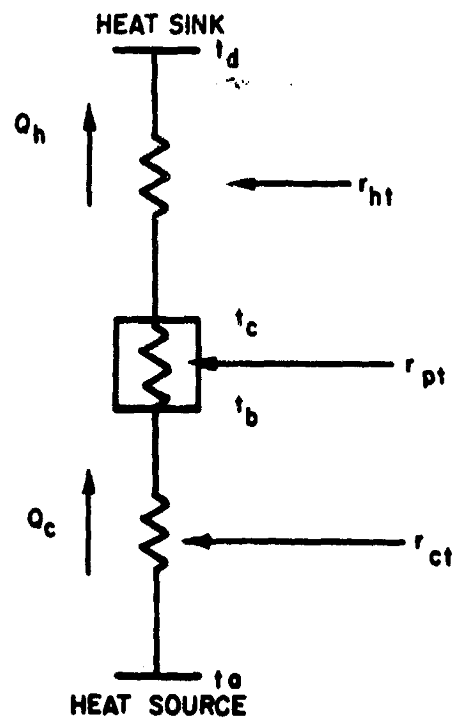


FIG. 3-1

THERMAL SCHEMATIC DIAGRAM OF A THERMOELECTRIC DEVICE

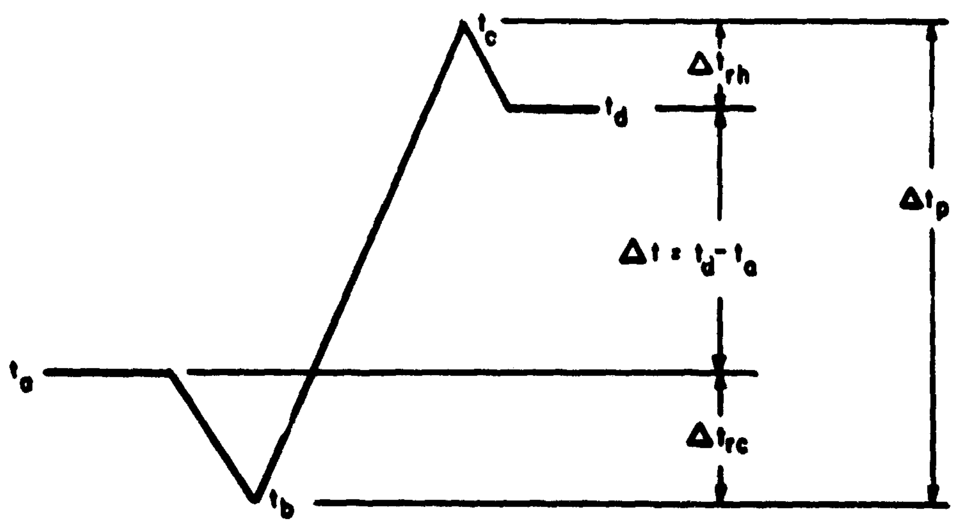


FIG. 3-2

THERMOELECTRIC DEVICE TEMPERATURE RELATIONSHIP

For purposes of this discussion, the temperature of the heat source (t_a) and the heat sink (t_d) are the average temperatures of the inside and outside air paths respectively. The average is derived from air inlet temperature and the temperature at the outlet of the heat exchangers.

The quantity of heat pumped (Q_c) must pass through the thermal resistance r_{ct} which is due to the conduction and film drops in the transfer path between the inside air and the cold junction. The temperature at the cold junction is t_b.

In the thermoelectric process a temperature difference must be produced across the thermoelectric which is sufficient to cause the rejected heat (Q_h) to flow through the hot side heat exchangers (thermal resistance r_{ht}) to the outside air at temperature t_d. Therefore, the hot junction temperature (t_c) is the sum of $t_d + \Delta t_{rh}$.

The temperature difference Δt_p is equal to $\Delta t_{rc} + \Delta t_{rh} + \Delta t$;

where Δt_p = temperature difference between the hot and cold junctions

Δt_{rc} = temperature difference across the thermal resistance at the cold junction

Δt = temperature difference between the average inside and outside air

Δt_{rh} = temperature difference across the thermal resistance at the hot junction. $(t_d - t_a)$

Referring again to equation (1) it will be noted that an increase in Δt_p will cause a reduction in the heat removed from the heat source. The magnitude of the effect of Δt_p on the power requirement for a given heat pumping capacity (24,000 Btu/hr) is shown by Figure 3-3. The Coefficient of Performance is also shown as a function of Δt_p.

With given outside and inside air inlet temperatures, the designer may exercise a limited choice of Δt_p by the selection of parameters such as air flow volume and heat exchanger geometry since both affect the three components of Δt_p, namely, Δt, Δt_{rc} and Δt_{rh}.

THERMOELECTRIC
POWER AND MAX. COP
VS
PELLET TEMP. DIFFERENCE

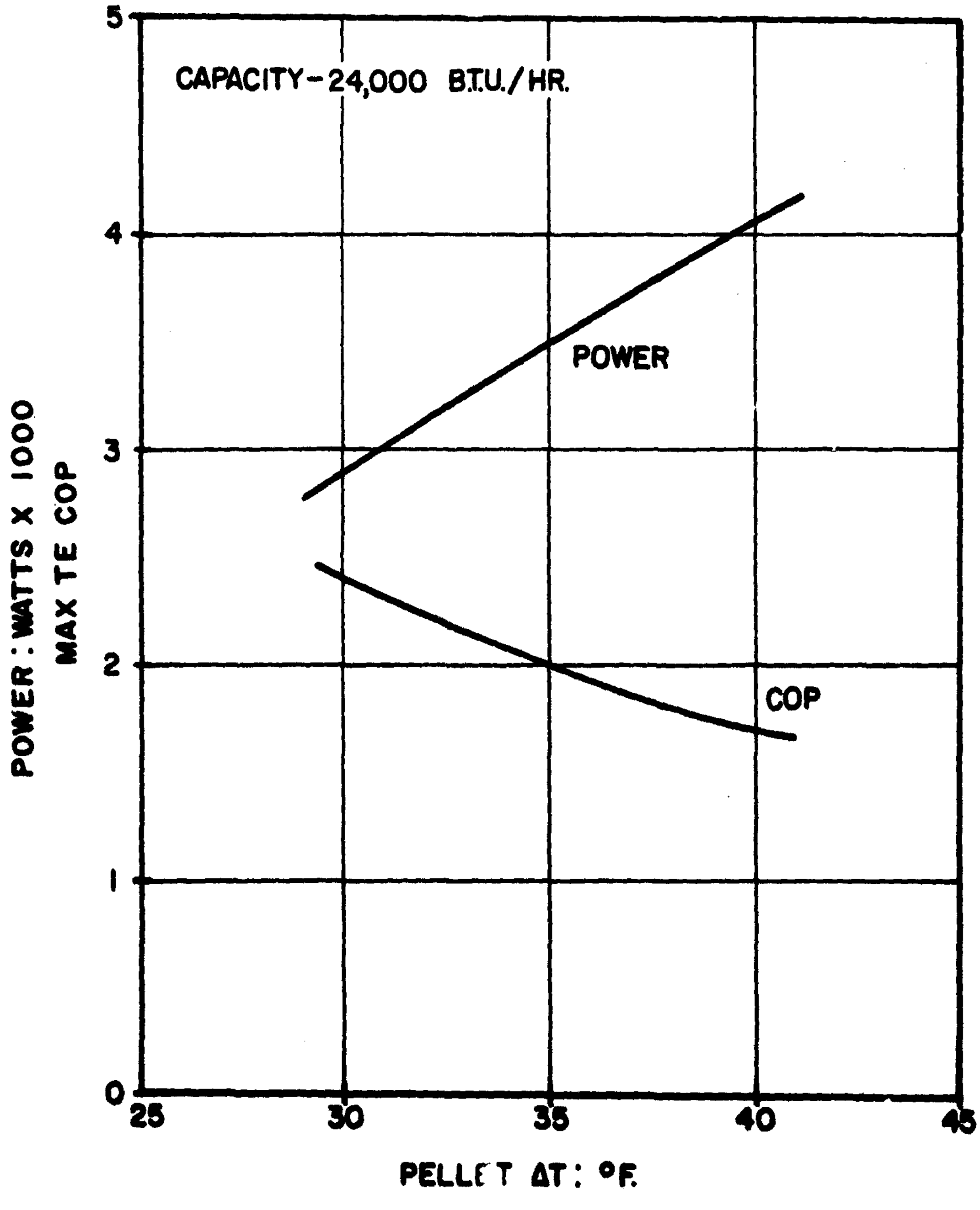


FIG. 3-3

3.3.2 System Performance

The system performance is determined from the total power required to deliver rated cooling capacity. It was noted from the preceeding discussion that the power for a given heat pumping capacity could be decreased by effecting a decrease in Δt_p. System performance optimization involves the determination of the gross heat pumping load, auxiliary power (fan power) and other system losses which also vary as air velocity, heat exchanger geometry, etc. are altered.

The gross heat pumping load is a summation of the following:

1. Net capacity (24,000 Btu/hr)
2. Heat load due to friction resulting from flow of inside air through its heat exchangers.
3. Heat leakage between the bases of the inside and outside air heat exchangers.
4. Heat transfer between the inside and outside air ducts.
5. Joule heat losses in the connecter between T/E pellets at the cold junction.
6. Inside air fan losses including motor.

The power to drive the inside and outside fans falls into the auxiliary power category and represents approximately 20% of the total system power. Because of its relatively large magnitude, fan power is one of the principal factors in performance optimization.

Considering the outside air fan operating at a fixed rate, the fan power will increase as the length of flow through the heat exchangers is increased. This relationship is represented graphically in Figure 3-4. The minimum power to operate the fan would exist when the flow length is a minimum.

For a given length of flow through the heat exchangers, the power necessary to operate, say the outside air fan will increase as the flow rate increases. However, as the flow rate increases, the power necessary to operate the thermoelectric core decreases. The

optimum condition exists at the flow rate at which the sum of the fan power and core power is a minimum. This condition is represented by point "A" in Figure 3-5.

Several other losses are included in the total power evaluation. They are:

1. Power supply losses
2. Resistance losses (I^2R losses) external to the T/E pellet.

Performance optimization involves the determination of the actual value of each power component. They are highly interrelated and a function of air velocity, heat exchanger geometry and package arrangement. The complex analysis is performed with the aid of digital computer programs.

3.4 Conventional vs. Direct Transfer Module Construction

3.4.1 Description

Two basic forms of modules were investigated for construction of the thermoelectric core, namely, "Conventional" and "Direct Transfer" designs. The conventional sub-module design is illustrated in Figure 3-6. The pellets are arranged in one plane and connected alternately "P" and "N" type by connecting straps. The connecting straps are bonded to insulation sheets which insulate the straps from the heat exchangers. The connectors result in all the junctions on one side being cold junctions and those on the other side being hot junctions.

The principal problems that must be overcome in the conventional module are related to the thermal losses through the insulation, thermal shock and cost. Good electrical insulators are normally poor thermal conductors. Small gaps, fractions of mils, also represent substantial thermal resistance. In order to obtain high performance with the conventional type, it is necessary to use high density ceramic insulating sheets that are metallized and bonded metallurgically to the connecting strap and heat exchangers. Since the ceramic insulation is of a brittle nature, exceptional care has to be taken to limit the

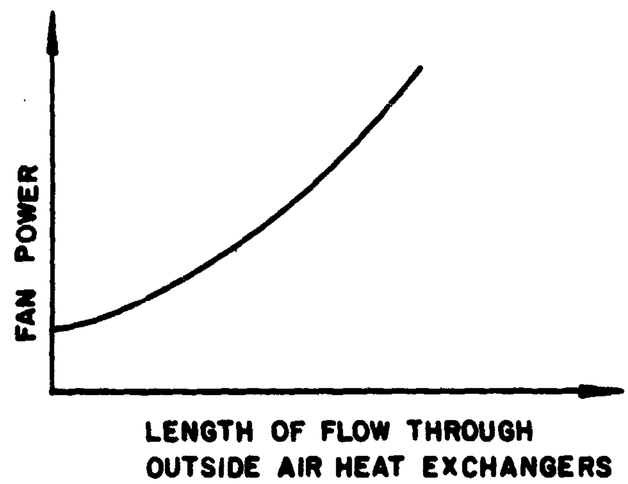

FAN POWER
LENGTH OF FLOW THROUGH
OUTSIDE AIR HEAT EXCHANGERS

FIG. 3-4

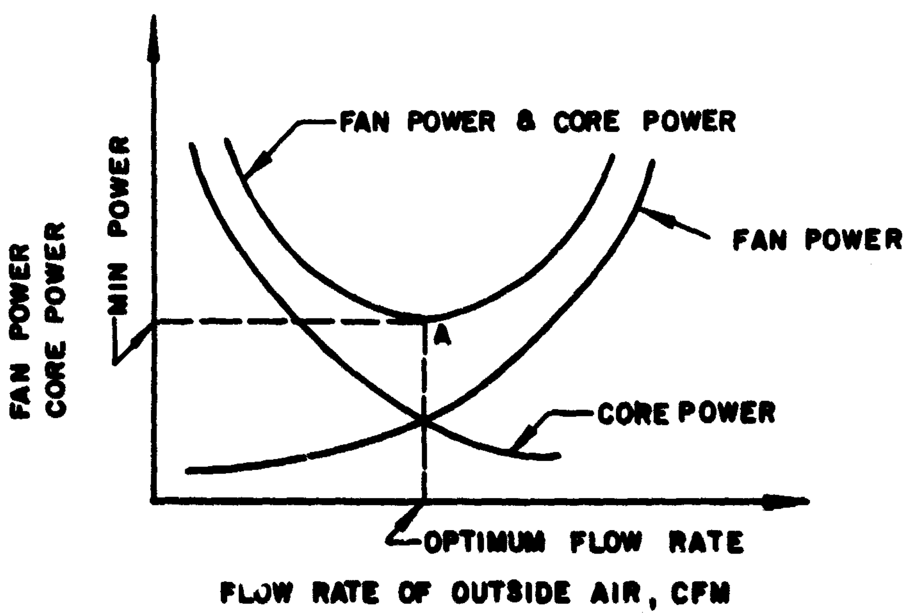
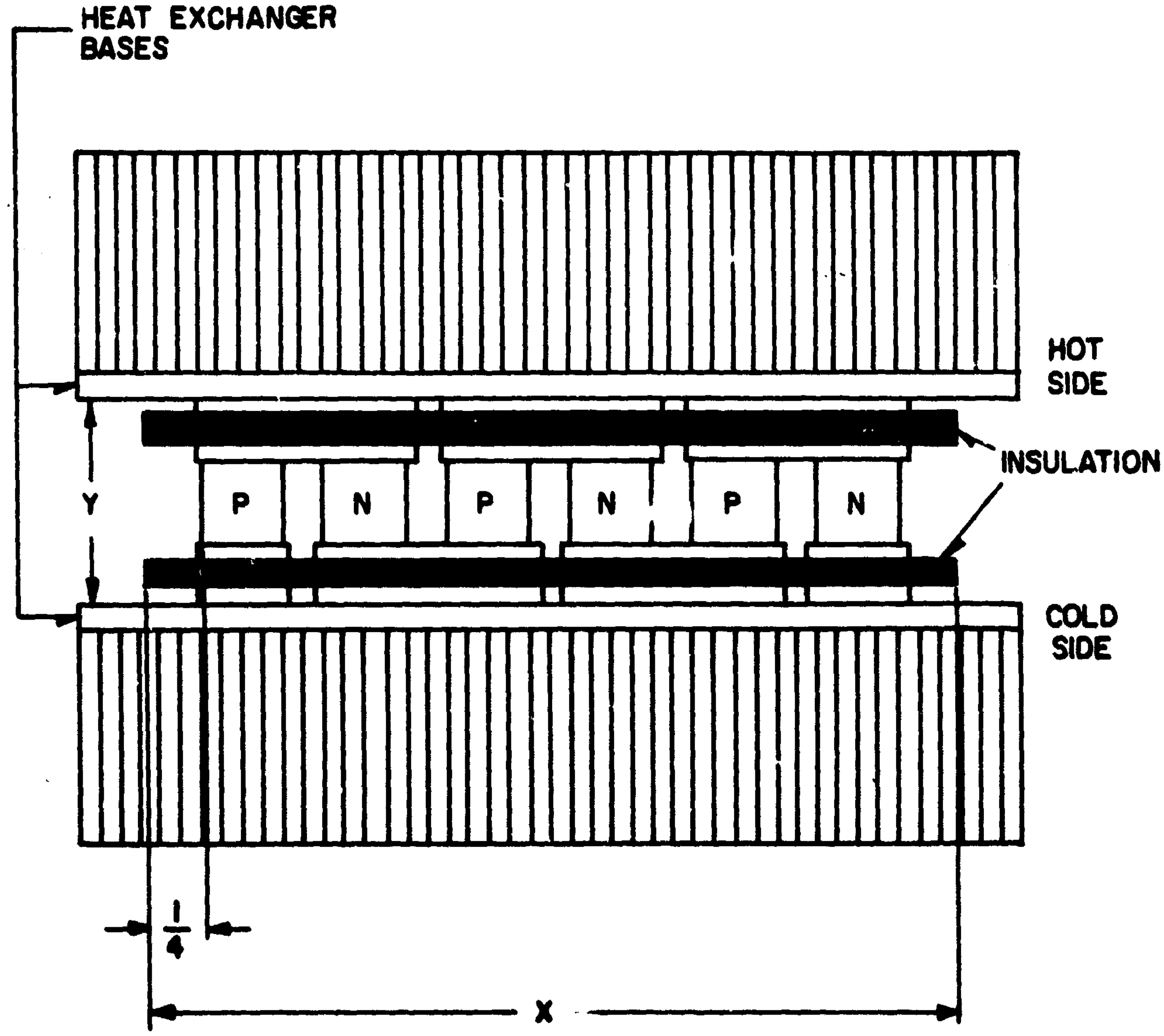

FAN POWER
CORE POWER
MIN POWER
FAN POWER & CORE POWER
FAN POWER
A
CORE POWER
OPTIMUM FLOW RATE
FLOW RATE OF OUTSIDE AIR, CFM

FIG. 3-5



CONVENTIONAL SUB-MODULE

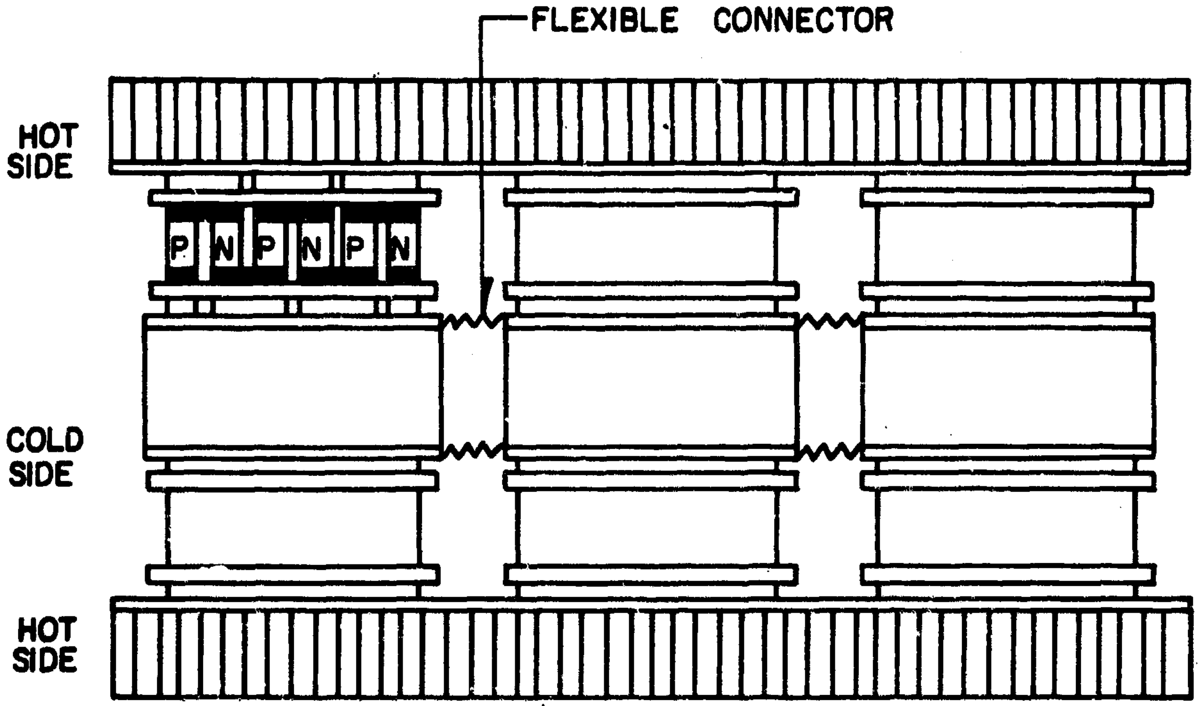
FIG. 3-6

stresses caused by thermal expansion. It is also important to prevent moisture from penetrating the ceramic insulation which would lead to low insulation resistance, high current leakages and eventual breakdown.

The sub-module considered for the conventional design used the Beryllia insulation system. Figure 3-6 is typical of an 18 couple sub-module. The insulation was extended 1/4" beyond the straps to provide electrical creepage. The dimensions of the sub-module are Y = .5" by X = 2 1/2" square.

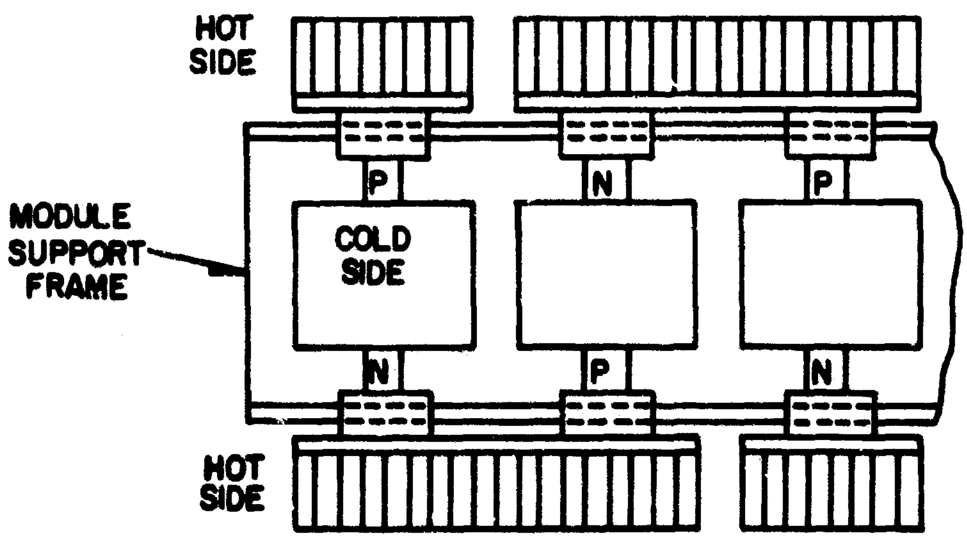
Approximately 24 sub-modules would be combined to form a module with a net rating of 2000 Btu/hr. The module construction technique is illustrated by Figure 3-7. Two modules are sandwiched between alternate hot and cold junction heat exchangers. The hot side heat exchangers perform a dual function in that they also provide mechanical support for the assembly. An individual heat exchanger is shown on the cold side for each pair of sub-modules. The relatively small cold side heat exchangers are used to limit the stress build-up due to differential expansion between the hot and cold sides.

For comparison, Figure 3-8 on the same page illustrates the construction of the "Direct Transfer" module. The "Direct Transfer" device has been evolved with the object of removing the electrical insulation from the heat transfer path. A module consists of an assembly of heat exchangers, alternately hot and cold planes with alternate P and N pellets sandwiched between each pair of hot and cold heat exchangers. The heat exchangers act as current carrying conductors and the pellets are metallurgically bonded to the heat exchangers as indicated. The fact that electric current flows through the fins of the heat exchangers does not lead to any difficulty. The voltage drop between fins or between heat exchangers is so low that current leakage, even under fouling conditions, is negligible.



CONVENTIONAL MODULE

FIG. 3-7



DIRECT TRANSFER MODULE

FIG. 3-8

3.4.2 Evaluation

With respect to other system components, the principal influence of the two basic forms of modules is on the power supply. Since each pellet in the "Direct Transfer" system is associated with a separate heat exchanger, economics demands that the system be constructed with relatively few pellets.

The effect of the number of pellets on power supply characteristics is illustrated by Figure 3-9. (This curve is based on a gross heat pumping capacity of 28,000 Btu/hr and t_p of 35°F which is a design condition used for comparison purposes) The simplest power converters, either a 3 phase half wave or a 3 phase full wave bridge will convert 208 volt, 3 phase, AC power to 140 volts DC or 280 volts DC respectively without introducing hardware for voltage transformation. For operation at 140 volts, 30 amp, DC, the system would require approximately 9500 pellets arranged in electrical series. At 280 volts, 15 amp DC the quantity of pellets is doubled. (19,000)

To realize the economic and reliability advantages of the "Direct Transfer" system the number of pellets must be minimized. This calls for a transformation to a relatively low voltage high current system. Considering power supply efficiency decreasing as a function of voltage, current conductor sizes along with I^2R losses, and the use of a standard module in smaller capacity units; 200 amperes at 20 volts DC was selected for the "Direct Transfer" approach. This system requires approximately 1300 pellets. The substantial reduction in pellets represents an exponential improvement in realiability of the current path.

In summary, the decision lies between the use of the "Conventional" module which is relatively independent of voltage along with its power supply and the "Direct Transfer" module and its associated low voltage power supply.

Using the principal factors selected for system evaluation, the following comparisons were made:

NUMBER OF PELLETS AND OPERATING VOLTAGE VS. OPERATING CURRENT

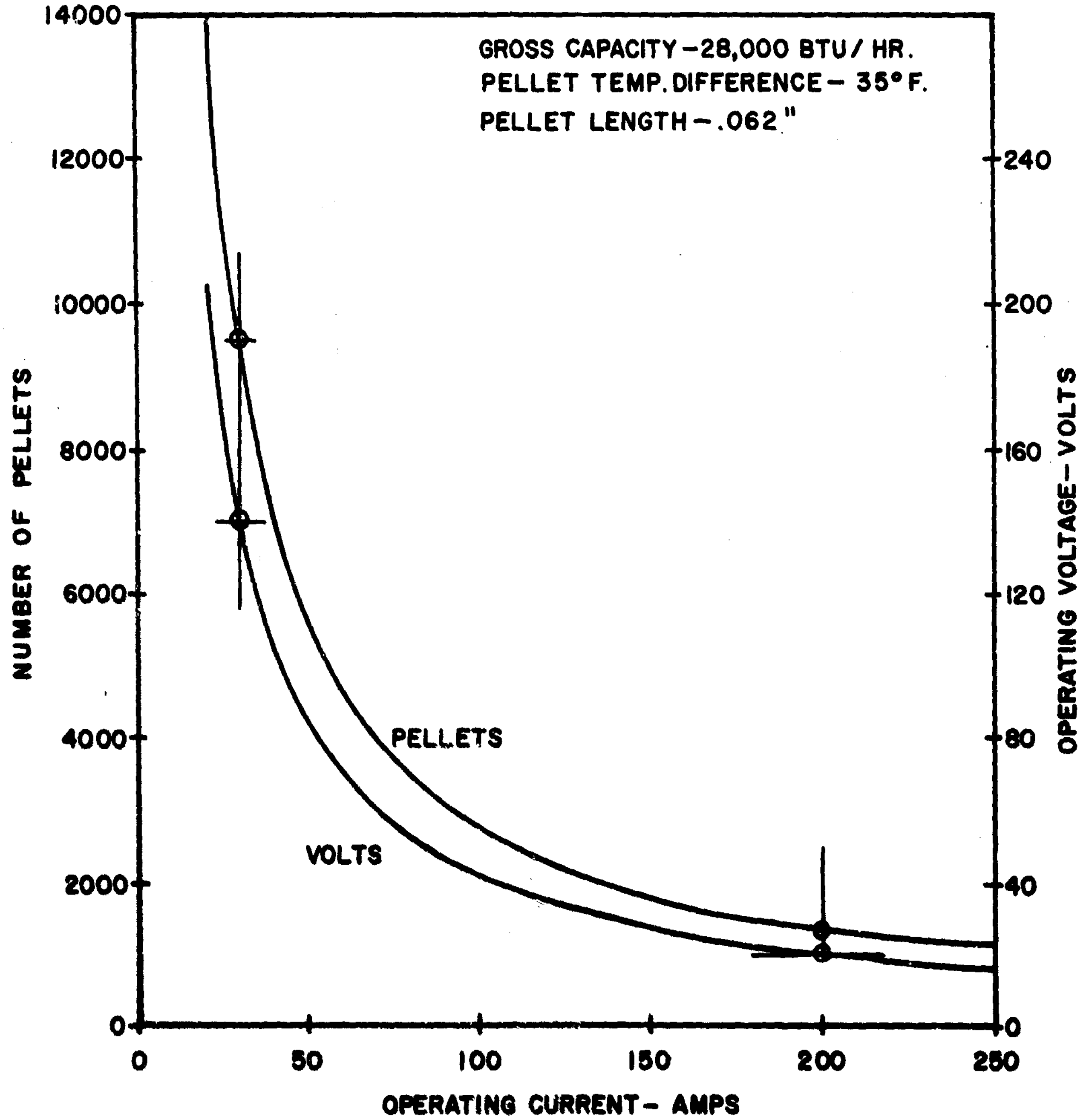


FIG. 3-9

1. Weight: The "Conventional" system weight exceeded the Direct Transfer System weight by 55 to 65 lbs.
2. Volume: Because of packaging limitations, the same core volume was used for both arrangements. This, however, resulted in a 10% reduction in heat exchanger surface area in the "Conventional" arrangement because of the greater thickness of the sub-module.
3. Performance: The pellet temperature difference and the resulting thermoelectric power increased in the "Conventional" design because of the module thermal resistance and the reduced heat exchanger area. This power increment is negated by the higher loss in the low voltage power supply used with the "Direct Transfer" design. Therefore, the COP of the two are nearly the same at full load.
4. Reliability: Based on experience to date, the life of "Conventional" modules which are metallurgically joined to both hot and cold heat exchangers has been quite brief. The failures have occurred due to thermal cycling and moisture penetration. The more advanced Beryllia system shows promise of marked improvement, but experience is very limited.

   This is contrasted with "Direct Transfer" systems build by Westinghouse Electric Corporation with which no failures or degradation in performance has been experienced. System construction was preceeded with thermal cycling tests which established 60,000 sudden 80°F thermal shocks as a minimum life with no degradation in performance.

5. Cost: System cost estimates were developed for the two designs based on buys in quantities of 600 units. The cost advantage for the "Direct Transfer" system varied between $2700 and $3900 per system. The rather large spread in the cost increment stemmed from the uncertainty in the quantity cost of the "Conventional" sub-module and the module construction technique which would withstand thermal cycling.

3.4.3 Conclusion

Because of the significant reliability and cost factors, further optimization and final design was based on the "Direct Transfer" concept.

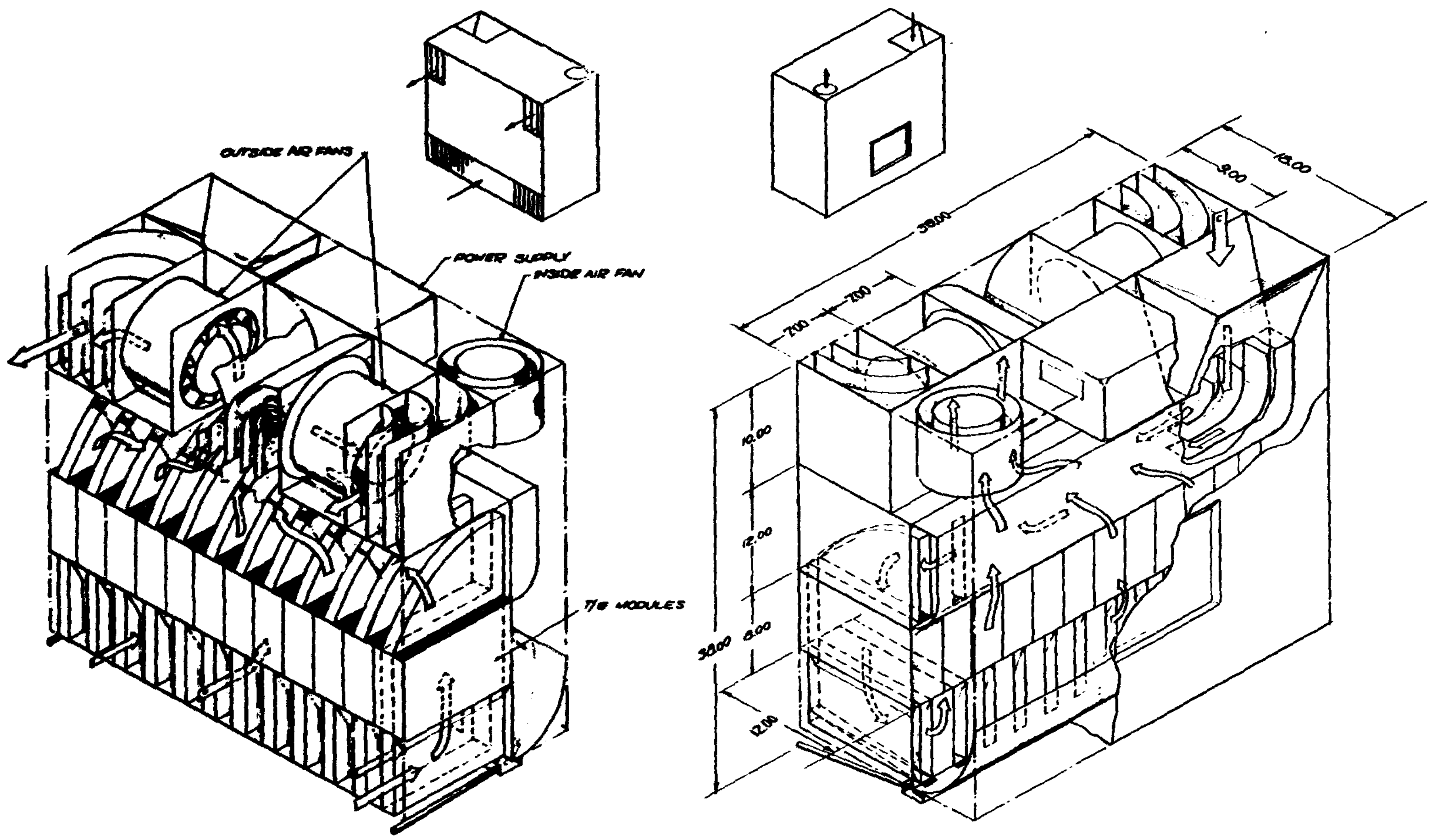
3.5 Crossflow - Counterflow Studies

Crossflow and counterflow of the air through the thermoelectric core were considered in detail in this study. The volume and dimensions of the core and the complete unit were determined for both types of flow by considering the overall performance of the total system in conjunction with the maximum allowable package dimensions.

A counterflow design, which was proposed as a possible design for this contract, is illustrated in Figure 3-10. The crossflow design is illustrated in Figure 3-11. The similarity in the dimensions of each drawing should be noted.

The core volume and the core dimensions are the same for both designs. The type of fin and the fin height used for the inside and outside air heat exchangers were also identical for each design; therefore, the effective heat transfer areas were also the same.

The counterflow design consists of 12 modules and the crossflow design consists of 24 modules. Each crossflow module is half the volume and half the cooling capacity (heat pumped) of a counterflow module.



PRELIMINARY COUNTERFLOW DESIGN

FIG. 3-10

Westinghouse Electric Corporation
ATOMIC EQUIPMENT DIV — CHESWICK, PA. U.S.A.

T/E UNIT
AIR TO AIR TYPE

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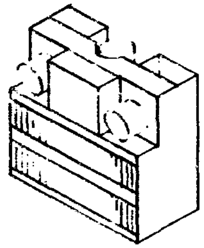
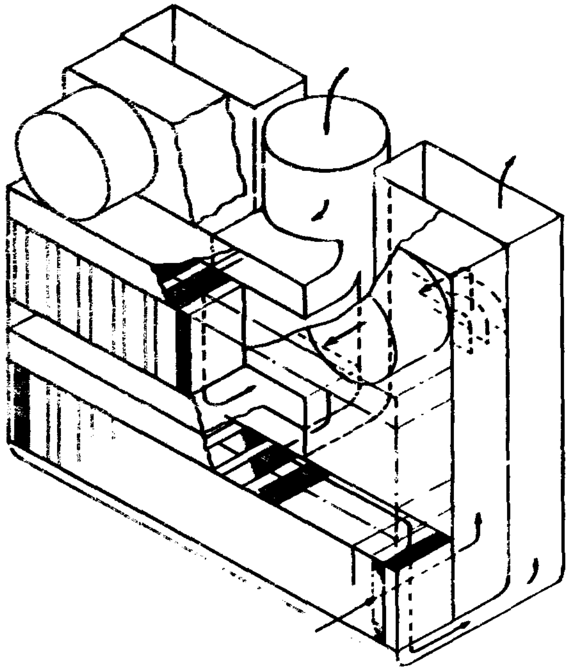
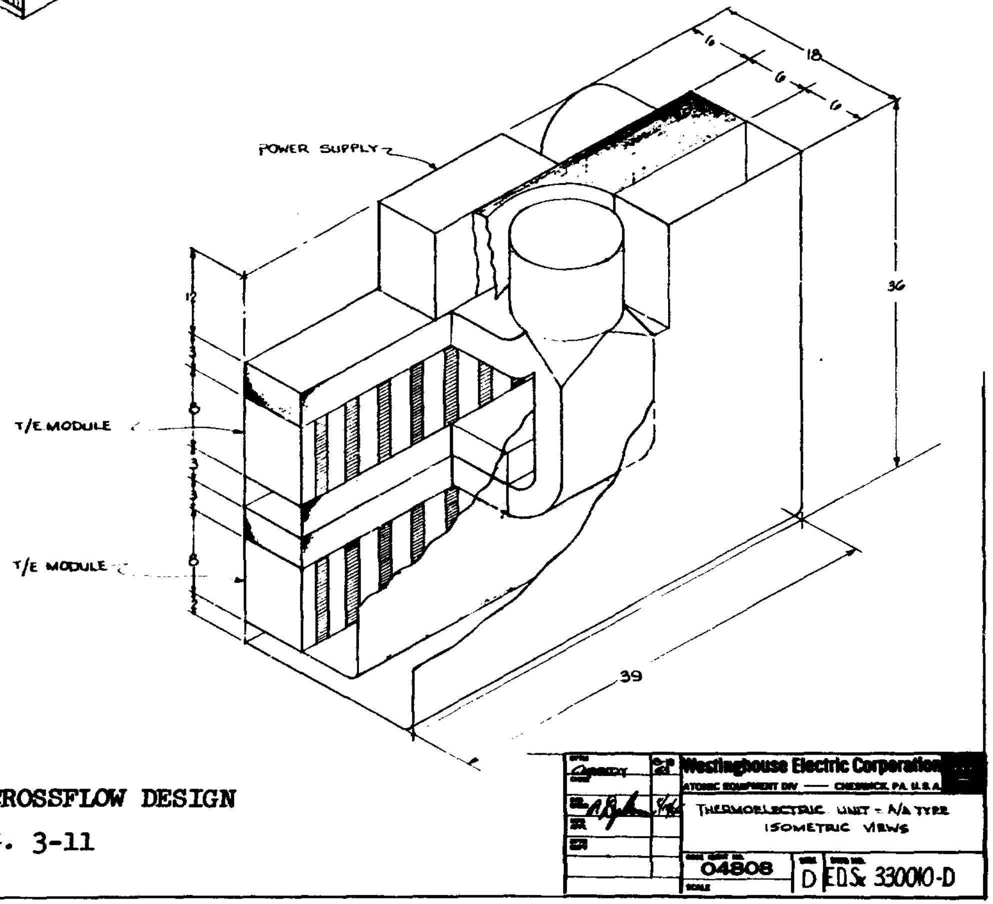






PRELIMINARY CROSSFLOW DESIGN

FIG. 3-11

The cross sectional area perpendicular to the inside air flow is the same for each design; however, in the crossflow design, the inside air flows through the core in two parallel branches. The cross sectional area for the outside air is not the same for both designs. For the crossflow design, the corss sectional area is 16/12 or 4/3 times the cross sectional area for the counterflow design.

The length of flow through the inside air heat exchangers is the same for both designs. For the outside air, the length of flow through the crossflow heat exchangers is 6 in. as compared to 8 in. for the counterflow design. Therefore, the crossflow heat exchanger flow length is 6/8 or 3/4 times the counterflow heat exchanger flow length.

Input fan power is equal to a dimensional constant times the product of the flow rate of the air and the system pressure drop divided by the fan efficiency. The pressure drop experienced by either the inside or outside air as it flows through the unit is equal to the sum of the pressure drops through the heat exchangers and the ducting system external to the core. The ducting system pressure drop experienced by the inside and outside air was approximately the same for both designs. For a fixed heat exchanger geometry with a fixed velocity of air passing through the heat exchangers, the pressure drop varies directly as the flow length. Therefore, for a fixed heat exchanger velocity and geometry, the input fan power for either the inside air or outside air varies directly as the product of the flow rate and flow length and inversely with the fan efficiency.

The flow rate of the inside supply air was specified at 840 to 1000 CFM. 840 CFM was considered the flow rate of the air because it would result in the best system performance. Since the cross sectional area of the inside air heat exchangers is the same for both designs, the velocity through the heat exchangers would also be the same (approximately 1000 FPM). Since the velocity, flow rate, and flow length are the same for both designs, the input fan power would also be the same.

The optimum velocity for the outside air was approximately the same for both designs (approximately 1100 FPM). To maintain this velocity through the outside air heat exchangers, the corssflow design would require 16/12 or 4/3 times the flow rate that would be necessary for the counterflow design. This is due to the difference between the cross sectional areas of the two designs. As previously mentioned, the crossflow heat exchanger flow length is 3/4 times the counterflow heat exchanger flow elngth. Therefore, at the optimum velocity for the outside air, the pressure drop through the crossflow heat exchangers would be 3/4 of the pressure drop through the counterflow heat exchangers. Assuming that the fan efficiency for both designs are the same, the power input to each fan would therefore be the same.

It can be concluded that the total input fan power would be approximately the same for both designs. However, the mass rate of flow of the outside air for the crossflow design would be 33-1/3% greater than the mass rate of flow for the counterflow design. For a fixed amount of heat pumped to the outside air, the temperature differences between the inlet and outlet of the outside air is inversely proportional to the mass rate of flow through the heat exchangers. For a fixed inlet outside air temperature and a fixed heat pumping rate to the outside air, the outlet temperature will be 25% lower for the crossflow design than for the counterflow design. Therefore, the average outside air temperature will be 25% lower for the crossflow design than for the counterflow design. This will result in a lower average temperature difference across the thermoelectric pellets for the crossflow design. The core COP will therefore be greater for the crossflow design.

In order to maintain the same pressure drop through the ducting system of the inside and outside air for both designs, it would be necessary to vane the inlet and outlet inside air ducts and the inlet outside air ducts for the thermoelectric core of the counterflow design. In Figure 3-10, these ducts were not drawn vaned for clarity.

Heat leakage between the outside air and the inside air exists in both designs. The leakage area of the crossflow design is approximately one half that of the counterflow design. Heat leakage represents an additional cooling load which must be removed by the core in order to supply inside air from the unit at a specified temperature. Therefore, the required cooling load of the crossflow design will be lower than that of the counterflow design.

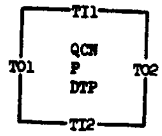
The performance of the thermoelectric core was analyzed by considering the average pellet in the core. The temperature difference between the inside and outside air for this average pellet was considered the difference between the average outside air temperature and the average inside air temperature of the air passing through the core.

If the power to the core is fixed and the temperature difference between the inside and outside air is the same at each pellet, then the performance of the core based on the average pellet will be the same as the actual performance of the core. However, if the temperature difference between the inside and outside air varies throughout the core, the performance is different than that based on an average pellet.

For a counterflow system having the increase of the outside air equal to the decrease of the inside air, the average pellet analysis would be correct. This is because the temperature difference across each pellet would be the same. For a crossflow system, the temperature difference across each pellet would vary throughout the core.

The difference between the performance based on a average pellet and the actual performance of a crossflow system was investigated in this study. The crossflow core illustrated schematically in Figure 3-11 was used for this analysis. Each pellet in the core was analyzed individually with the aid of a computer program to determine its operating characteristics. The results of this study are schematically represented in Figure 3-12. Based on the same net cooling capacity, the total power input to the core determined from this analysis was 5% greater than the total power input calculated by using an average pellet.

CROSSFLOW PERFORMANCE ANALYSIS



TI1 - INSIDE AIR INLET TEMPERATURE, °F
TI2 - INSIDE AIR OUTLET TEMPERATURE, °F
TO1 - OUTSIDE AIR INLET TEMPERATURE, °F
TO2 - OUTSIDE AIR OUTLET TEMPERATURE, °F
QCN - PERCENT OF TOTAL HEAT PUMPED, %
P - PERCENT OF TOTAL POWER INPUT, %
DTP - TEMPERATURE DIFFERENCE ACROSS A PELLET, °F

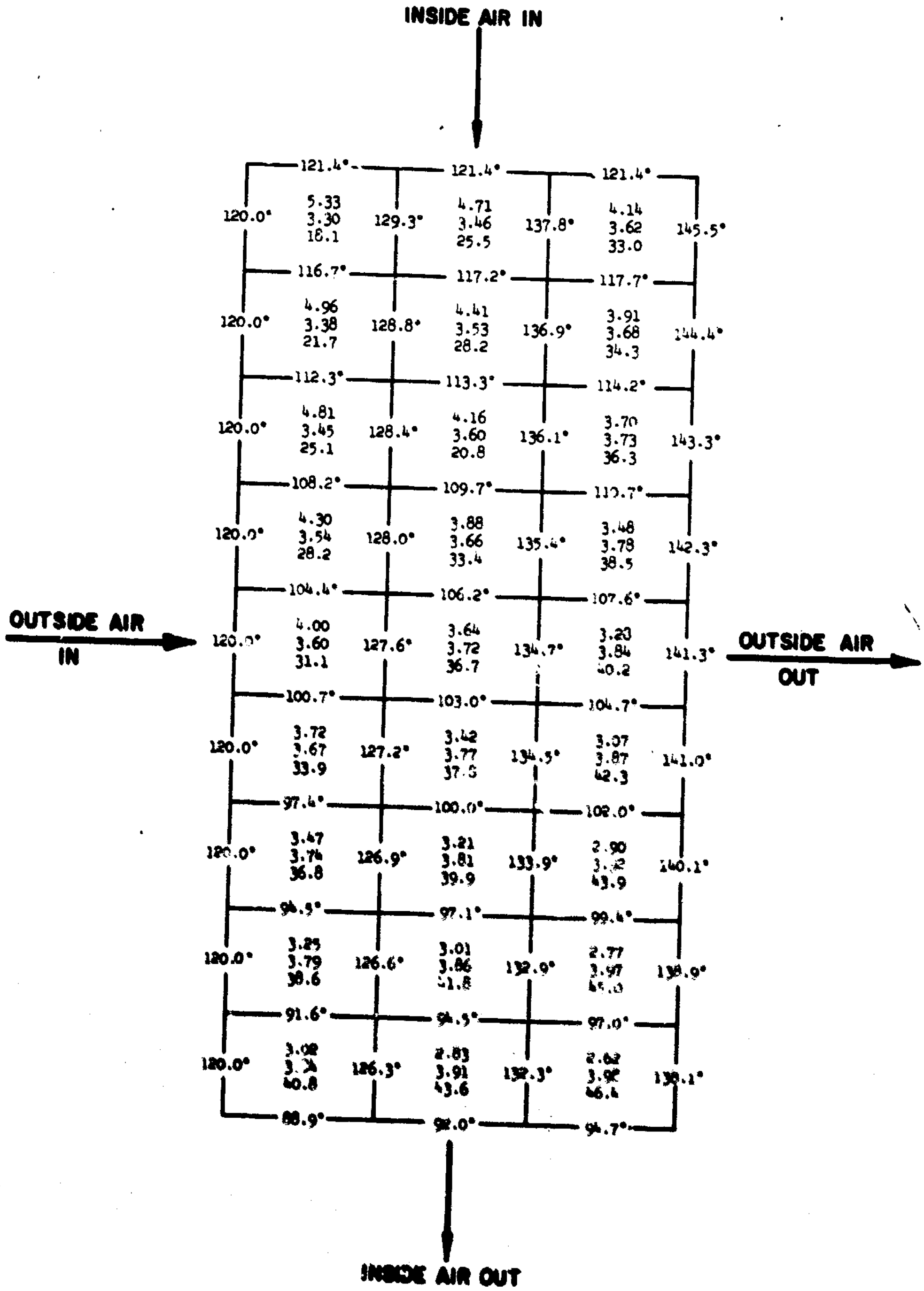


FIG. 3-12

Considering the counterflow design (Figure 3-10) with the same inlet inside and outside air temperature as used in the crossflow analysis (Figure 3-11), the temperature increase of the outside air would not equal the temperature decrease of the inside air as the air travels through the core. The temperature increase of the inside air would be approximately 8°F greater than the decrease in the outside air. It is estimated that the total power input to the core calculated by analyzing each pellet individually would be approximately 1% greater than the total power input calculated by considering an average pellet.

Below is a summary of the input power difference between the crossflow design and the counterflow design. Plus (+) power means that the input power to the counterflow design is greater than that of the crossflow design while a minus (-) power has the opposite meaning.

	Power Difference - Watts
Inside air fan	0
Outside air fan	0
Power difference due to mass flow of outside air	+600
Heat leakage	+175
Additional power due to pellet temperature difference variation	-150
Total power	+625 Watts

It can be concluded from this summary that on a performance basis the crossflow design is superior to the counterflow design.

The mechanical construction of the crossflow design and the counterflow design should be considered. It can be seen from Figures 3-10 and 3-11 that the ducting system for the inside and outside air is much simpler for the crossflow design. The vaned inlet and outlet ducts of the counterflow design have been eliminated. The air in the crossflow design enters and leaves the core in large plenum chambers. The ducting system of the crossflow design is approximately 15-20 lb. lower in weight than that of the counterflow design. The simplicity of the crossflow design will result in a lower unit cost than that of the counterflow design.

From the discussion of the crossflow-counterflow studies, it can be concluded that the crossflow design is superior to counterflow design. The crossflow method of air flow through the core was considered the preferred method for this contract.

3.6 Crossflow System Optimization

3.6.1 Heat Exchangers

The design of a compact heat exchanger is quite complicated. The performance of a heat exchanger is dependent upon the fin configuration used. The principal factors to be considered are heat transfer, pressure drop through the fins, weight, volume, condensate drainage, noise level, resistance to fouling, electrical conductivity, and mechanical properties.

When considering heat transfer, the principal objectives are to obtain maximum heat transfer from a given volume with the lowest possible thermal resistance and weight. In order to obtain a good heat transfer performance from a heat exchanger, it is desirable to operate in the turbulent region of flow. This means that the fins of the heat exchanger must be designed to obtain turbulent flow by breaking up the laminar flow.

The ability of a finned heat exchanger to transfer heat can be analyzed by considering the heat transfer coefficient h, which has units of $Btu/hr\text{-}ft^2\text{-}°F$. For a given heat exchanger operating at a fixed velocity and a fixed temperature difference between the air stream and base of the fins, the amount of heat transferred through the fins varies directly as the heat transfer coefficient. Therefore, for maximum heat transfer, the maximum possible heat transfer coefficient should exist.

The heat transfer coefficient varies with the velocity through the heat exchanger. However, the pressure drop experienced by the air as it flows through a heat exchanger varies directly as the velocity. The optimum design of a heat exchanger is therefore a compromise between heat transfer and pressure drop.

Various fineed surfaces were considered for the heat exchangers of the thermoelectric core. These surfaces were compared by considering the following factors fixed: air velocity, air temperature, fin height, fin base area and dimensions, fin material, and heat transferred through the fins. The finned surface which was found to give the optimum performance was the strip-fin plate-fin which is sometimes referred to as a staggered strip-fin. This finned surface was considered the preferred design for the thermoelectric core.

The performance curves and the dimensions of the strip-fin plate-fin surface used in the core are illustrated in Figure 3-13. The performance curves were based on the curves presented by W. M. Kays and A. L. London in their book "Compact Heat Exchangers", second edition. The data is presented in the form of a heat transfer factor, $j = (h/G\ c_p)\ N_{pr}{}^{2/3}$, and a friction factor, f, plotted against the Reynolds Number, N_R, where

h = convective heat transfer coefficient, Btu/hr-ft^2-°F

G = Exchanger flow stream mass velocity, lb/hr-ft^2

c_p = Specific heat at constant pressure, Btu/lb-°F

N_{pr} = Prandtl Number, c_p/k, dimensionless

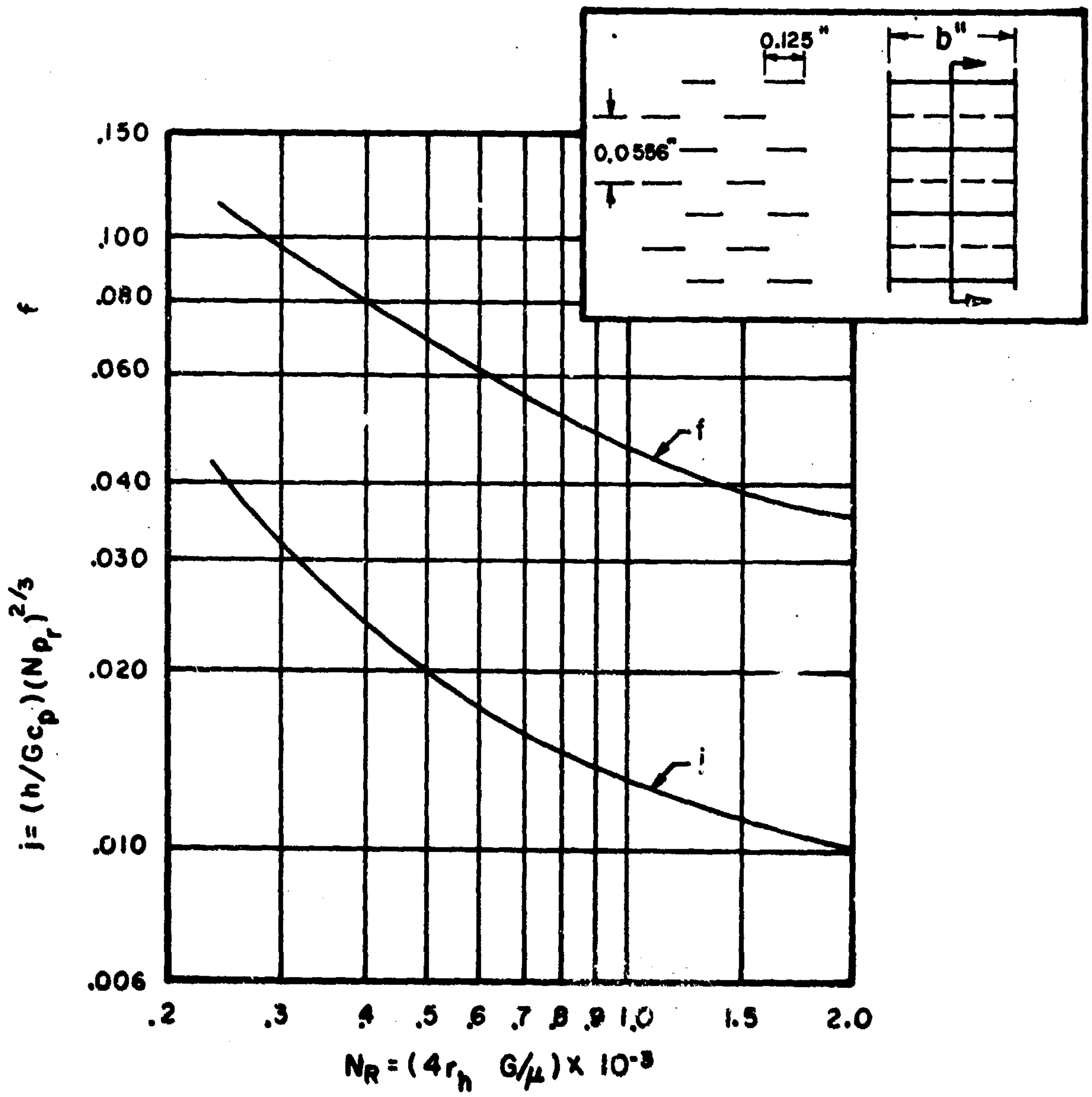
= Viscosity, lb/hr-ft

k = Thermal conductivity, Btu/hr-ft^2-°F/ft

r_h = Hydraulic radius, ft

3.6.2 Pellet Dimensions

The dimensions of the thermoelectric pellets, i.e., the length and cross sectional area, affects the performance of the thermoelectric core. This can be seen by examining equations (1), (2), and (3) in section 3.3.1. For a fixed current and fixed pellet hot and cold junction temperatures, the heat pumped, power input, and coefficient of performance of a thermoelectric couple are a function of the pellet area to length ratio (A/L).



STRIP-FIN PLATE-FIN

FIN PITCH = 18.00 PER IN.

PLATE SPACING, b IN.

FIN LENGTH FLOW DIRECTION = 0.125 IN.

FIN METAL THICKNESS = 0.010 IN., ALUMINUM

FIG. 3-13

The second term in equation (1) is joule heat loss of a couple and the third term is the heat leakage from the hot junction to the cold junction. Both terms detract from the heat pumping capabilities of a couple. Increasing the A/L ratio of a pellet, decreases the joule heat loss and increases the heat leakage. The selection of the optimum pellet dimensions is therefore a compromise between joule heat loss and heat leakage. The optimum pellet dimensions for a fixed current and fixed junction temperatures can be found by maximizing equation (3) with respect to A/L.

In Figure 3-14, pellet length is plotted against T/E material weight and heat pumping density. The curves are based on maximum COP at a 35°F temperature difference across the pellets. The following factors were considered fixed: heat pumping capacity, COP, current, hot and cold pellet junction temperature, and the number of pellets. The T/E material weight varies with the pellet length and the heat pumping density varies inversely with the pellet length

Because of the high cost of T/E material, it is desirable to use the smallest weight of material. This would correspond to the smallest practical pellet length. The pellet length which will be used for this contract is 1/16 in. From Figure 3-14, this corresponds to a total T/E material weight of approximately 9 lbs. Smaller pellet lengths have been used, but the added manufacturing difficulties do not warrant the slight savings in the total T/E material weight.

3.6.3 Optimization of the Core Dimensions

The optimum volume and dimensions of the core and the complete unit were determined by considering the overall performance of the total system within the limitations of the maximum allowable package dimensions and volume. A 24 module core was considered with 12 modules in the width direction. See Figure 7-7.

The fin heights of the inside and outside air heat exchangers were picked so that the total width of the core was approximately 38 in. This was done to keep the inside air heat exchanger flow length and the outside air heat exchanger flow length a minimum for a given core volume. The fin heights of the inside and outside air heat exchanger are 0.471 in. and 0.6875 in., respectively.

HEAT PUMPING DENSITY AND T/E MATERIAL
WEIGHT VS. PELLET LENGTH

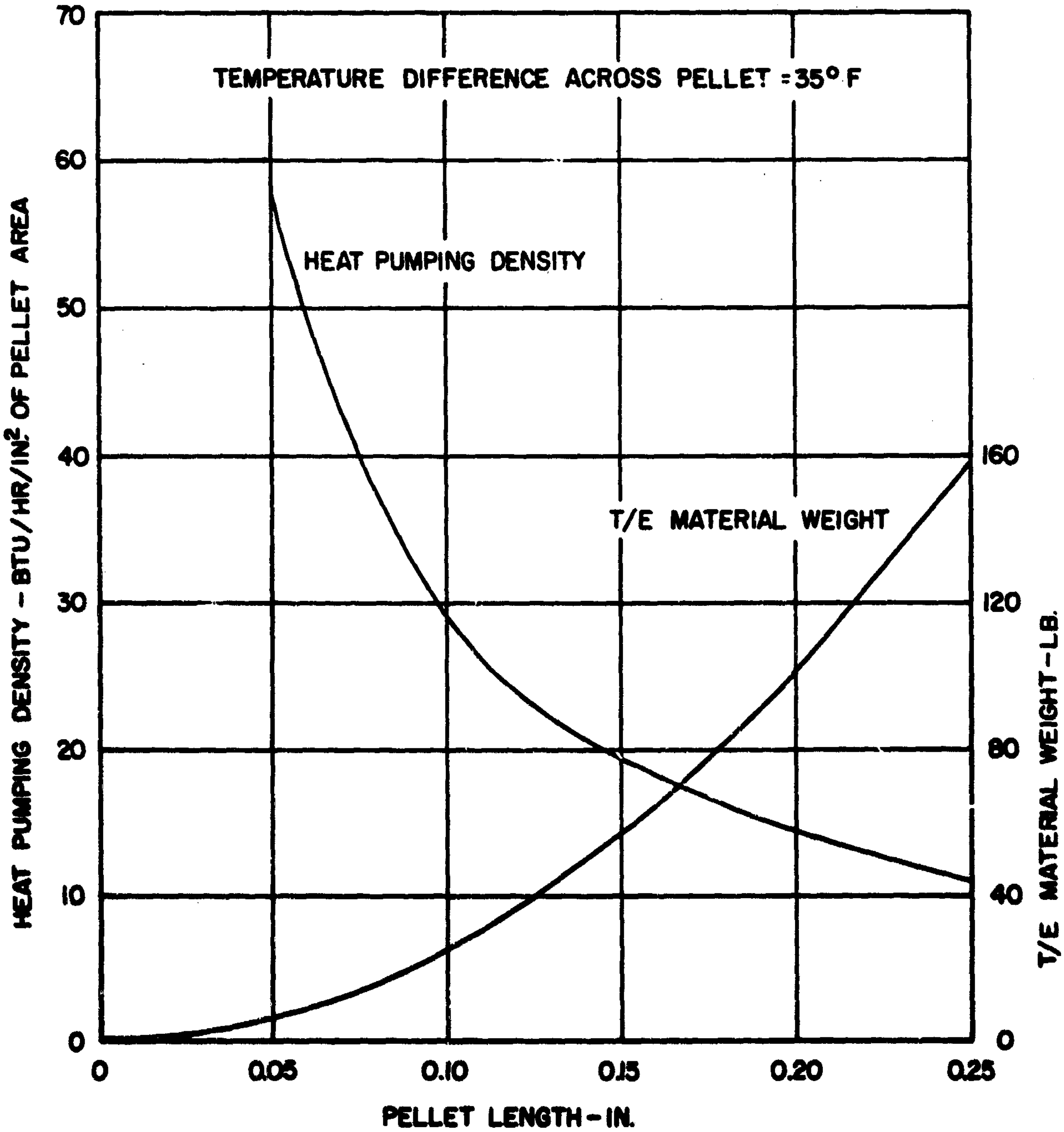


FIG. 3-14

For this combination of fin heights, it is necessary to know the optimum core dimensions. Since the flow rate of the inside air is fixed at 840 CFM, the inside air velocity through the core will vary directly as the outside air heat exchanger flow length. Since the flow rate of the outside air is not specified, the performance of the core and consequently the total system for fixed core dimensions will be dependent upon the flow rate of the outside air. Therefore, for fixed core dimensions, the core performance should be analyzed at various outside air flow rates.

The performance of the total system for various core dimensions and outside air flow rates was analyzed. The outside air flow length of a module was varied from 5 to 8 in. and the inside air flow length of a module was varied from 7 to 10 in. The ducting system pressure drop for the inside and outside air was considered constant for the various core configurations and outside air flow rates. The results of this study are illustrated graphically in Figures 3-15, 3-16, and 3-17. The outside air flow length is plotted against the inside air flow length for various system power inputs at outside air velocities of 800, 1000, and 1200 FPM.

For a fixed system power input, the outside air flow length varies inversely with the inside air flow length. For fixed core dimensions, fixing one flow length and increasing the other will result in a decrease in the system power input; however, fixing one flow length and decreasing the other will result in an increase in the system power input. A low system power input at a fixed outside air velocity will result from a core having large inside and outside air flow lengths or the combination of a large inside air flow length and a small outside air flow length. For either core geometry, the overall package volume and/or dimensions will exceed the unit specifications. For a fixed core geometry, the lowest system power input resulted from an outside air velocity of 1200 FPM.

The dotted lines in Figures 3-15, 3-16 and 3-17 represent the core dimensions which are considered to give the optimum system performance within the overall package limitations. The outside air flow length is 6.6 in. and the inside air flow length is 8.8 in. With these dimensions, a study was undertaken to determine whether an outside air velocity of 1200 FPM was the optimum velocity. The results of this study are represented graphically in Figure 3-18. The minimum system power input exists at an outside air velocity of approximately 1100 FPM. The difference between the system power input at 1100 and 1200 FPM is practically negligible. 1100 FPM was considered the outside air velocity. This corresponds to an outside air flow rate of 1800 CFM.

MODULE GEOMETRY
VS.
SYSTEM POWER INPUT

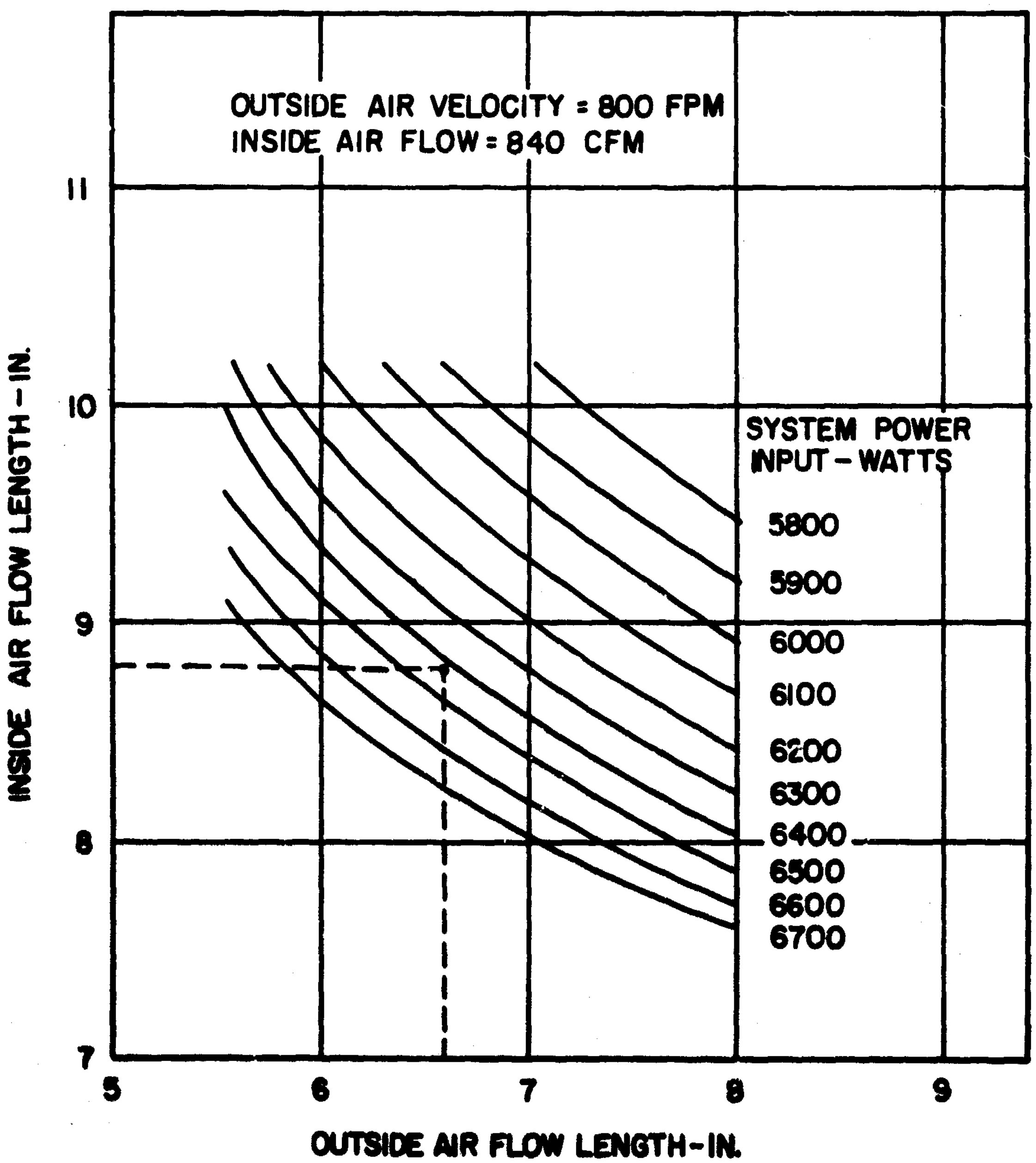


FIG. 3-15

MODULE GEOMETRY
VS.
SYSTEM POWER INPUT

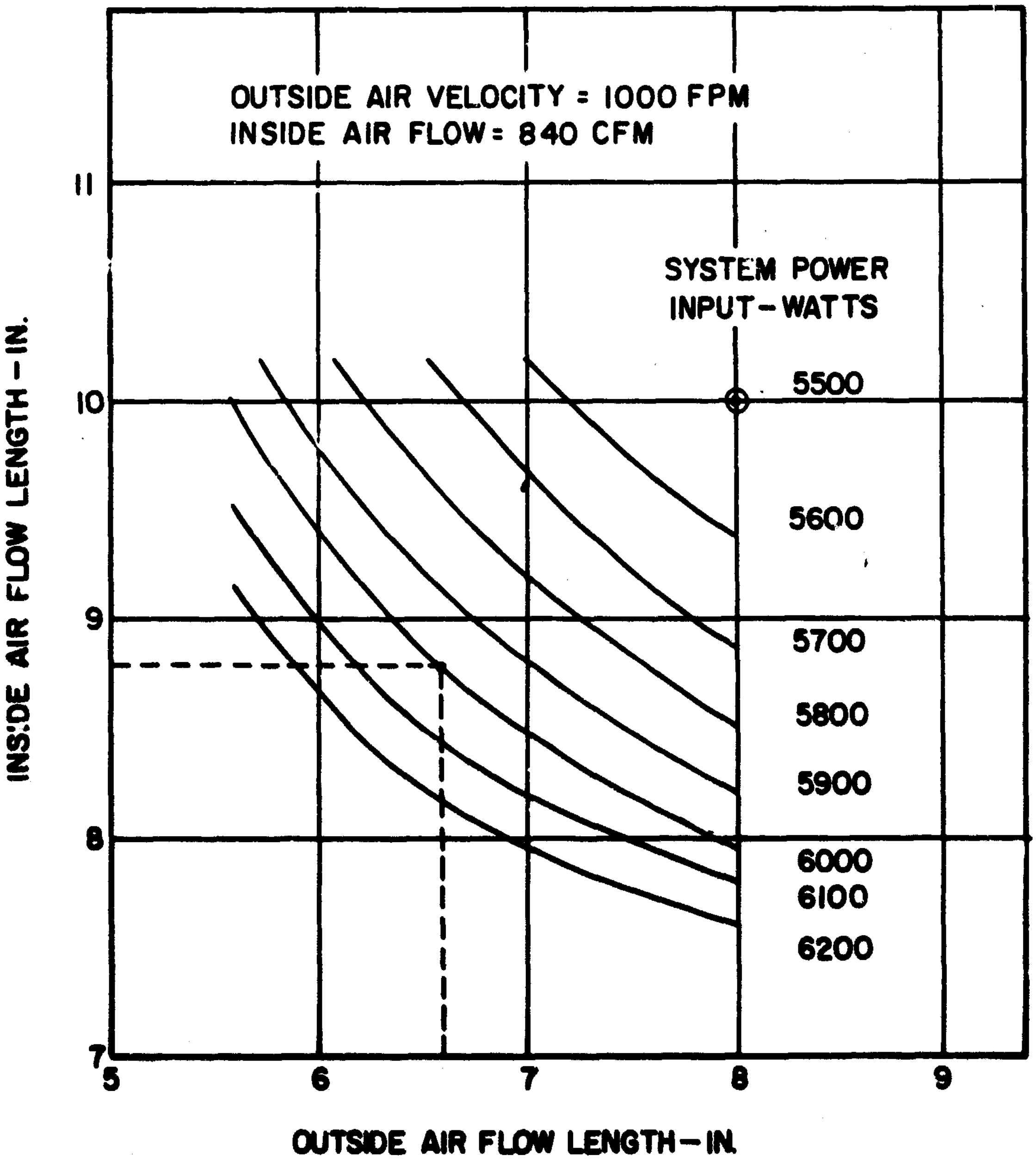


FIG. 3-16

MODULE GEOMETRY
VS
SYSTEM POWER INPUT

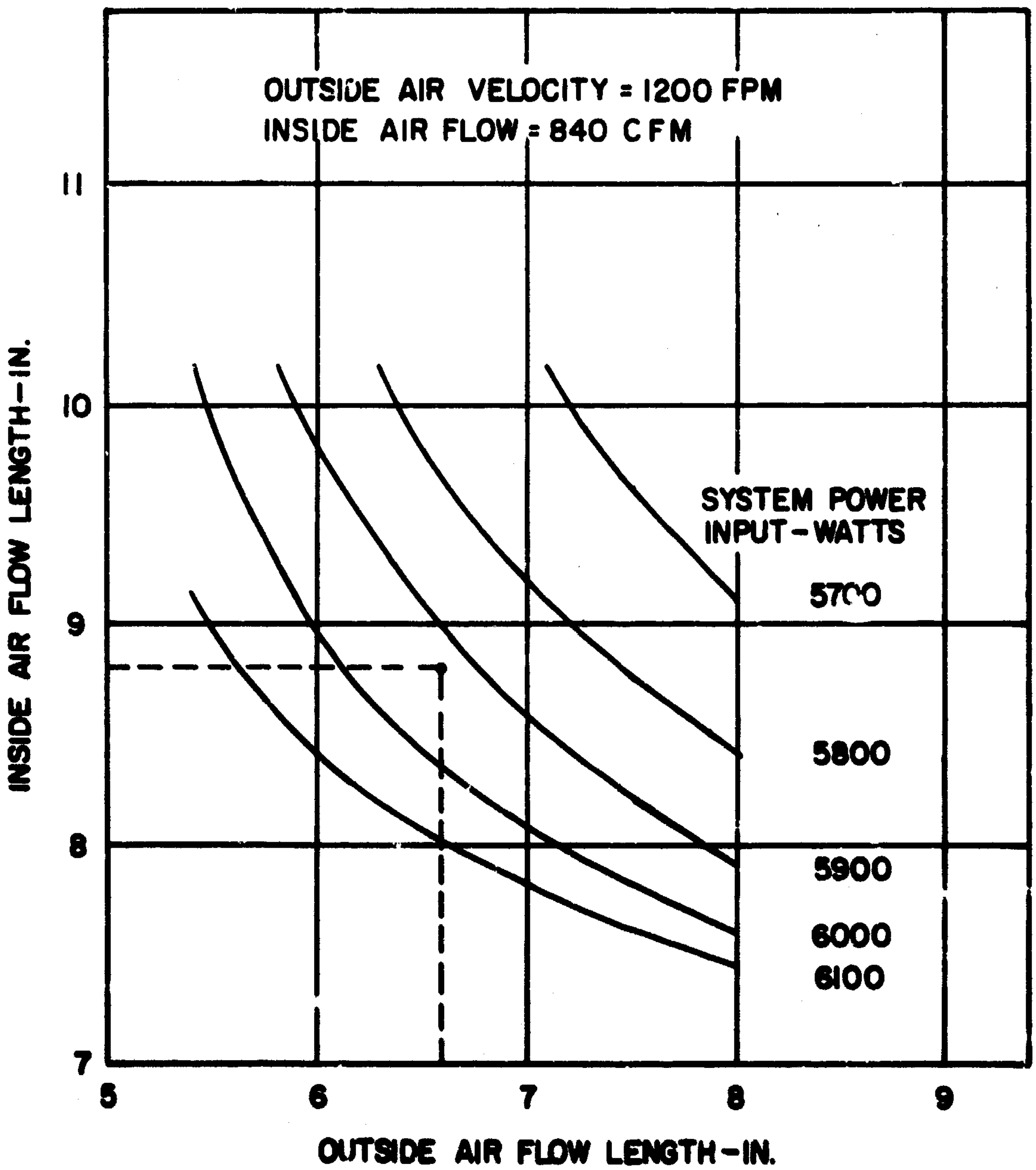


FIG. 3-17

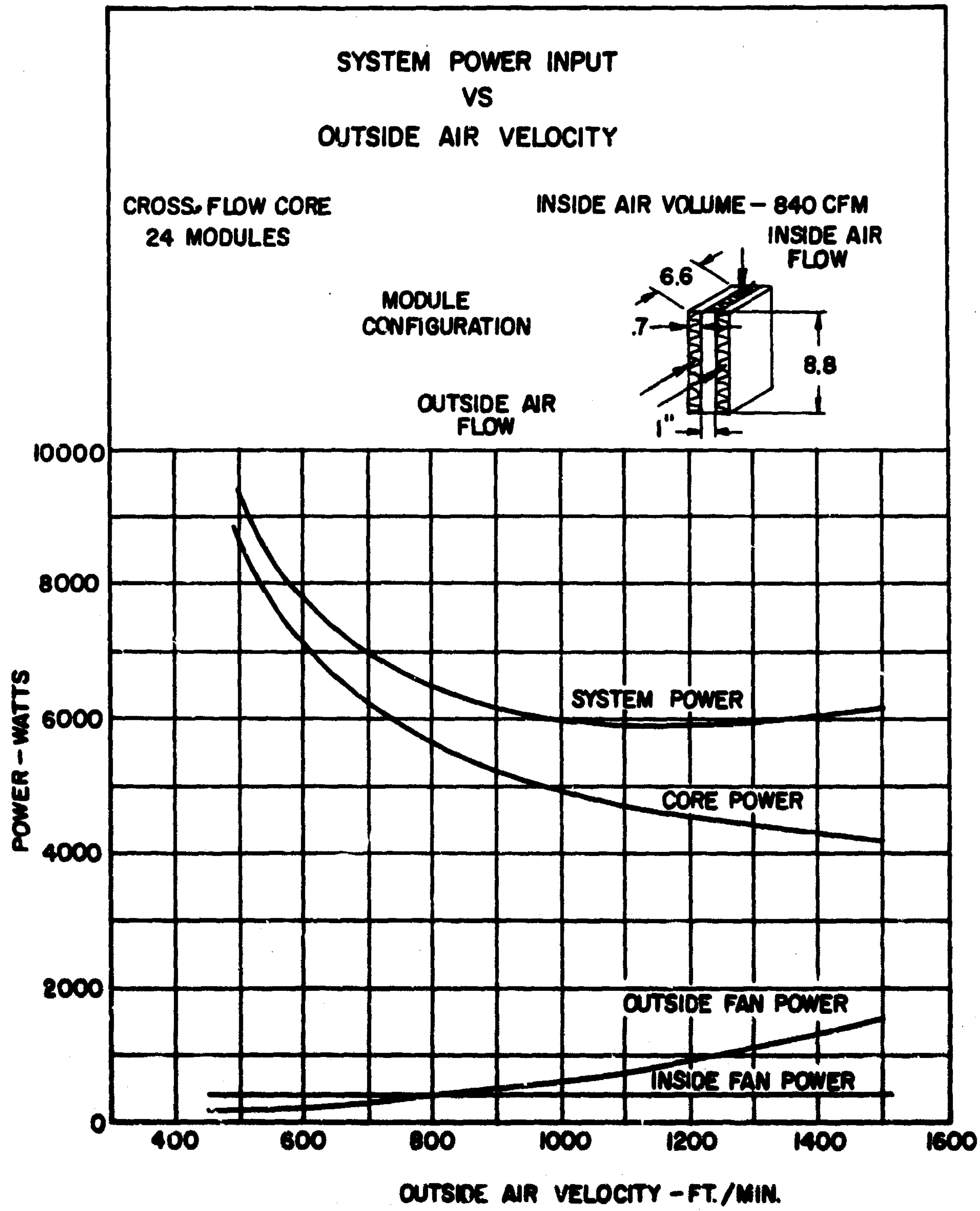

SYSTEM POWER INPUT
VS
OUTSIDE AIR VELOCITY
CROSS-FLOW CORE
24 MODULES
INSIDE AIR VOLUME - 840 CFM
INSIDE AIR FLOW
6.6
MODULE CONFIGURATION
.7
8.8
OUTSIDE AIR FLOW
1"
10000
8000
6000
4000
2000
0
POWER - WATTS
SYSTEM POWER
CORE POWER
OUTSIDE FAN POWER
INSIDE FAN POWER
400
600
800
1000
1200
1400
1600
OUTSIDE AIR VELOCITY - FT./MIN.

FIG. 3-18

4.0 ENVIRONMENTAL CONTROL UNIT DESIGN

4.1 General Unit Description

An integrated system of thermoelectric modules, solid state power supply, air moving equipment and appropriate ducting are combined as shown in Figure 7-7.

In the make-up of the thermoelectric unit twenty-four modules are connected in series. The module heat exchangers are oriented for crossflow of inside and outside air streams. One hundred eight individual pellets are sandwiched between aluminum heat exchangers in two parallel sheets to form a module.

Two outside air fans and power conversion capacity control packages are mounted directly above the thermoelectric cores.

The overall unit dimensions are 37-3/4 inches high, 38-1/8 inches wide and 18 inches deep. These dimensions envelope a volume of fifteen cubic feet with a unit dry weight of two hundred sixty-five pounds.

For Nominal Cooling Condition this compact, light weight unit produces 24,000 BTU per hour of cooling with a COP of 1.18. This COP value is based on 60 cycle operation as shown on Table I.

To achieve this heat pumping rate, 1800 cfm of outside air enters the outside wall in two horizontal, parallel paths, and passes through the grill, filter, thermoelectric modules and into the duct, where it is then turned vertically upwards. Here, the two parallel paths join and flow into channels at the sides of the duct. At the top of the duct, turning vanes are incorporated to guide the air into the fan entires. After going through the fans horizontally, the hot air is expelled through a wire mesh screen back out into the atmosphere.

At the same time 840 cfm of inside air flows vertically downward through the fan and then splits evenly into two paths. Turning vanes are used to guide the air into the sealed chambers above the thermoelectric modules. After passing out of the modules, collection chambers are used to direct the conditioned air to a plenum next to the inside wall of the unit. The cooled air is then exhausted vertically upward out one of the top corners depending on which side

the supply air is desired. Both sides are capable of being opened at the [illegible]e time if that is desired.

To handle condensate drainage, drain pans are located beneath each core. Holes are located on the outside wall to spill the water overboard from the unit. A small amount of inside air is bled out of the unit, but the effect on the performance of the unit is negligible.

The hot side air duct is drained by two holes in the bottom of the outlet duct. A plastic tube is used to connect the drain hole to the outside wall.

4.1.1 Unit Performance Pesults

Tables I and II give the specific operating conditions for different operating points which vary from Nominal Cooling to Nominal Heating. Figures 4-1, 4-2, 4-3, and 4-4 show the control of the unit with different ambient air conditions.

4.1.2 Unit Weight

Below is a weight estimate for individual components to arrive at a total system weight.

Thermoelectric Cores (24 modules)		124 lbs.
Structure		41 lbs.
3 Fans	18.5 lbs. each	56 lbs.
Power Conversion and Capacity Control		38 lbs.
Control and Monitor Panel		6 lbs.
TOTAL		265 lbs.

4.1.3 Unit Costs

Price estimates on the prototype and production quantities of 10, 50, and 600 units are covered under a separate transmittal letter.

4.2 COMPONENT SPECIFICATIONS

4.2.1 Structure

The structure is defined as the housing which holds the thermoelectric modules, fans, power supply and ducting. It is composed of two parts, the frame as shown in Figure 7-4 and the case. The structure

TABLE I

CONDITIONS	POINTS 1	2	3	4	5	6
Main Power Frequency (Cps)	60	60	60	60	60	60
Temp. Sensor Temp. (°F)	100	78	76	74	72	50
Supply Air Drybulb Temp. (°F)	92	72	66	68	72	50
Supply Air Wetbulb Temp. (°F)	63	60	58	54	50	0
Outdoor Ambient Drybulb (°F)	120	120	80	40	0	-50
Outdoor Ambient Wetbulb (°F)	69	69	60	35	0	-65
CFM Supply Air	840	839	794	794	839	765
Return Air Drybulb (°F)	120	81.0	92.0	76.0	106.0	63.0
CFM Outside Air	1800	1806	1674	1674	1535	--
Outside Air Leaving Module (°F)	141.8	132.3	97.6	86.4	58.9	--
Fan Power Heat Gain to Inside Air (Btu/hr)	750	758	667	673	758	605
Heat Conducted to Inside Air (Btu/hr)	550	1141	507	343	-372	-1170
Net Total Cooling Capacity (Btu/hr)	24,000	8080	21,926	8233	29,700	-11,285
Cold Fan Power (Watts)	440	441	391	393	441	336
Hot Fan Power (Watts)	789	792	688	683	579	--
Total Fan Power (Watts)	1229	1233	1079	1076	1020	336
Core Power & Power Supply Losses (K Watts)	4.760	4.970	2.630	0.604	0	3.990
Total Power (K Watts)	5.989	6.203	3.709	1.680	1.020	4.356
% Ripple	2	2	5	8	-	2
Net COP	1.18	0.382	1.73	1.44	8.53	0.76

TABLE II

CONDITIONS	POINTS 1	2	3	4	5	6
Main Power Frequency (Cps)	400	400	400	400	400	400
Temp. Sensor Temp. (°F)	100	78	76	74	72	50
Supply Air Drybulb Temp. (°F)	92	72	66	68	72	50
Supply Air Wetbulb Temp. (°F)	63	60	58	54	50	0
Outdoor Ambient Drybulb (°F)	120	120	80	40	0	-50
Outdoor Ambient Wetbulb (°F)	69	69	60	35	0	-65
CFM Supply Air	898	898	849	849	898	810
Return Air Dry Bulb (°F)	119.8	80.0	90.0	76.0	102.0	62.4
CFM Outside Air	1927	1932	1792	1792	1642	--
Outside Air Leaving Module (°F)	140.4	130.2	96.4	86.3	56.6	--
Fan Power Heat Gain to Inside Air (Btu/hr)	736	743	502	657	743	562
Heat Conducted to Inside Air (Btu/hr)	852	1095	723	316	-408	-1158
Net Cooling Capacity (Btu/hr)	23,912	7692	21,941	8124	28,365	-11,254
Cold Fan Power (Watts)	469	472	418	418	472	357
Hot Fan Power (Watts)	842	843	732	732	614	--
Total Fan Power (Watts)	1311	1315	1150	1150	1086	357
Core Power & Power Supply Losses (K Watts)	4.750	4.950	2.640	.681	0	3990
Total Power (K Watts)	6.061	6.265	3.790	1.831	1.086	4345
% Ripple	2	2	5	8	-	2
Net COP	1.15	0.359	1.70	1.30	7.65	0.76

NET COOLING VS APPLIED VOLTAGE

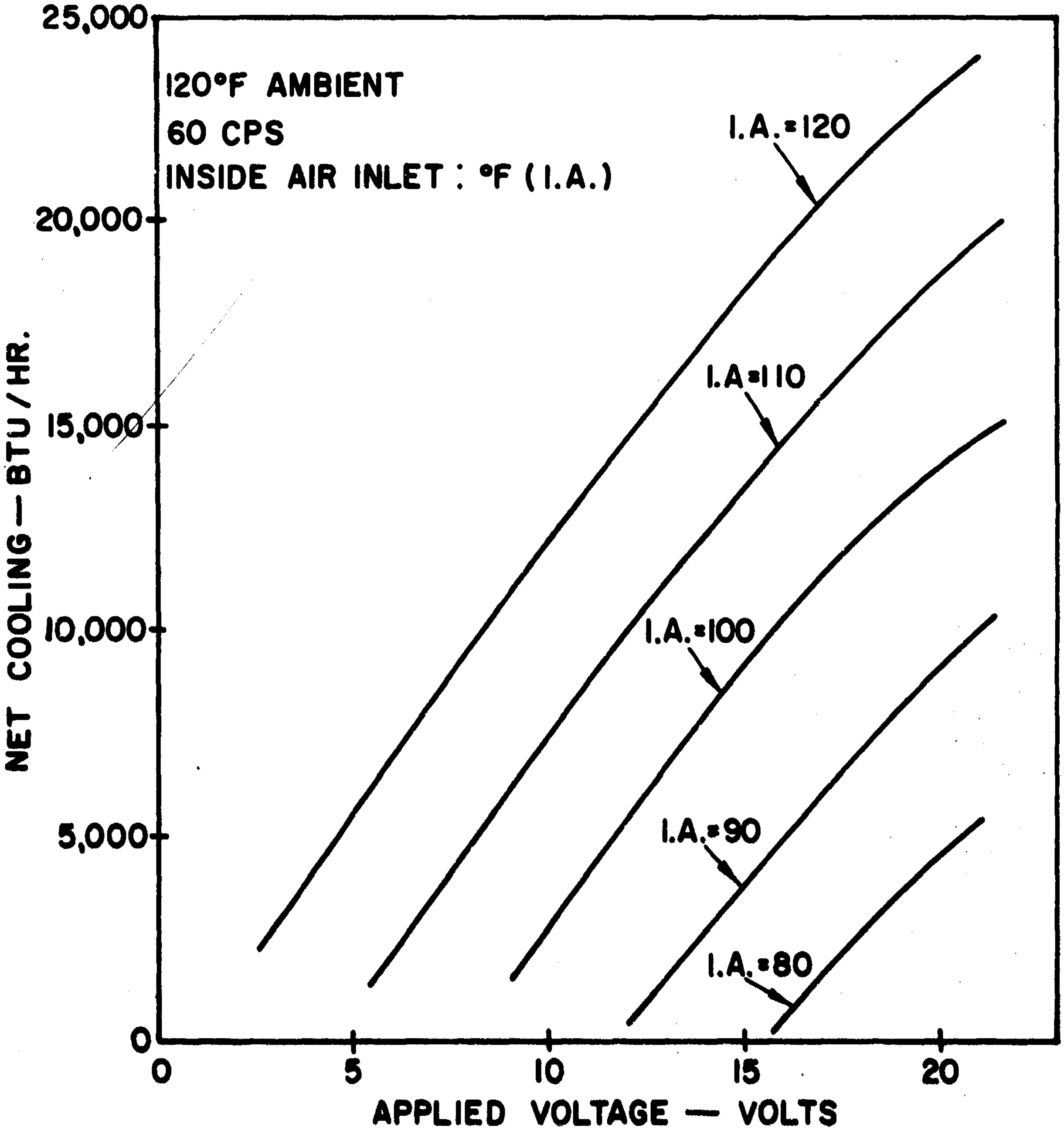


FIG. 4-1

NET COOLING VS APPLIED VOLTAGE

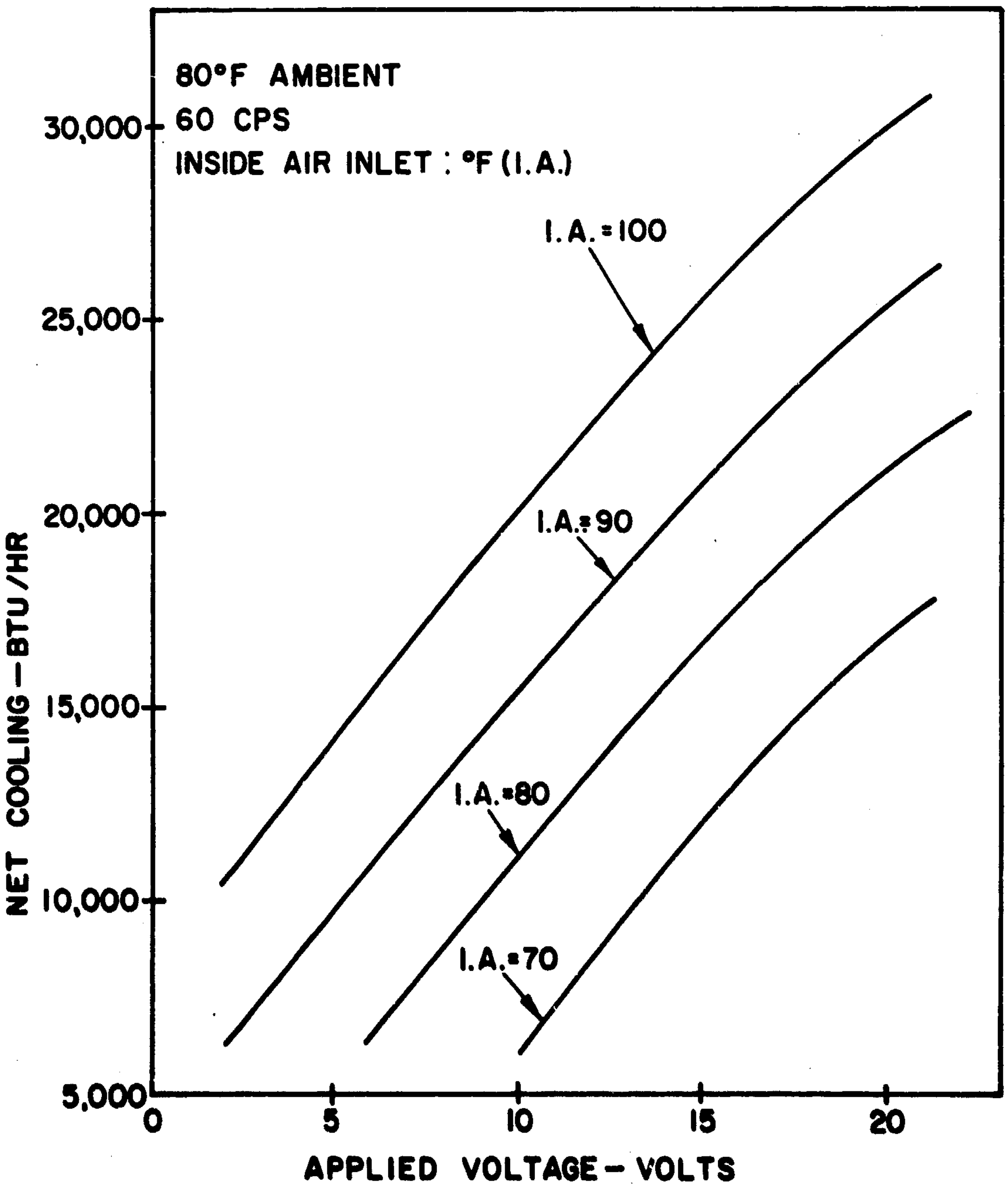


FIG. 4-2

NET COOLING VS APPLIED VOLTAGE

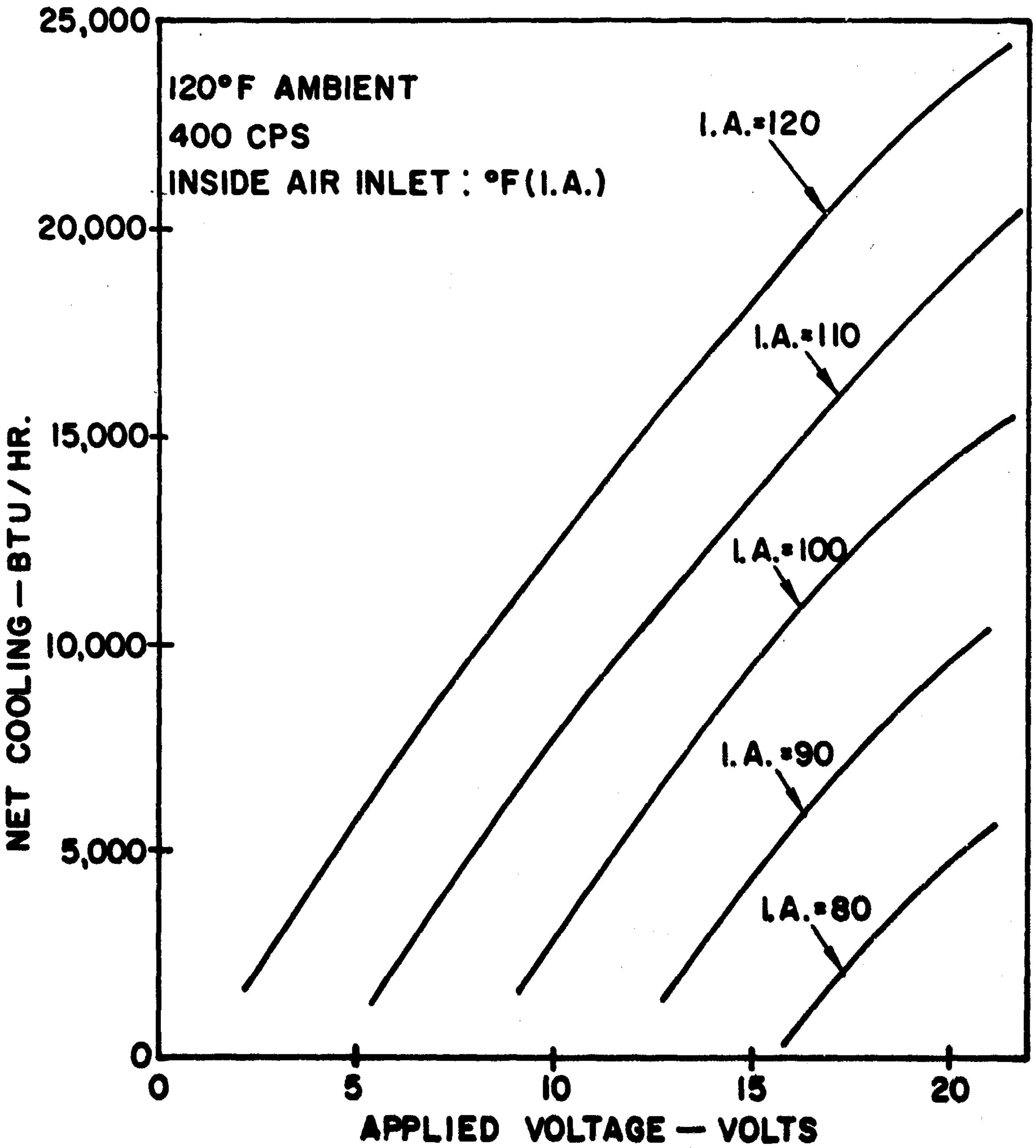


FIG. 4-3

NET COOLING VS APPLIED VOLTAGE

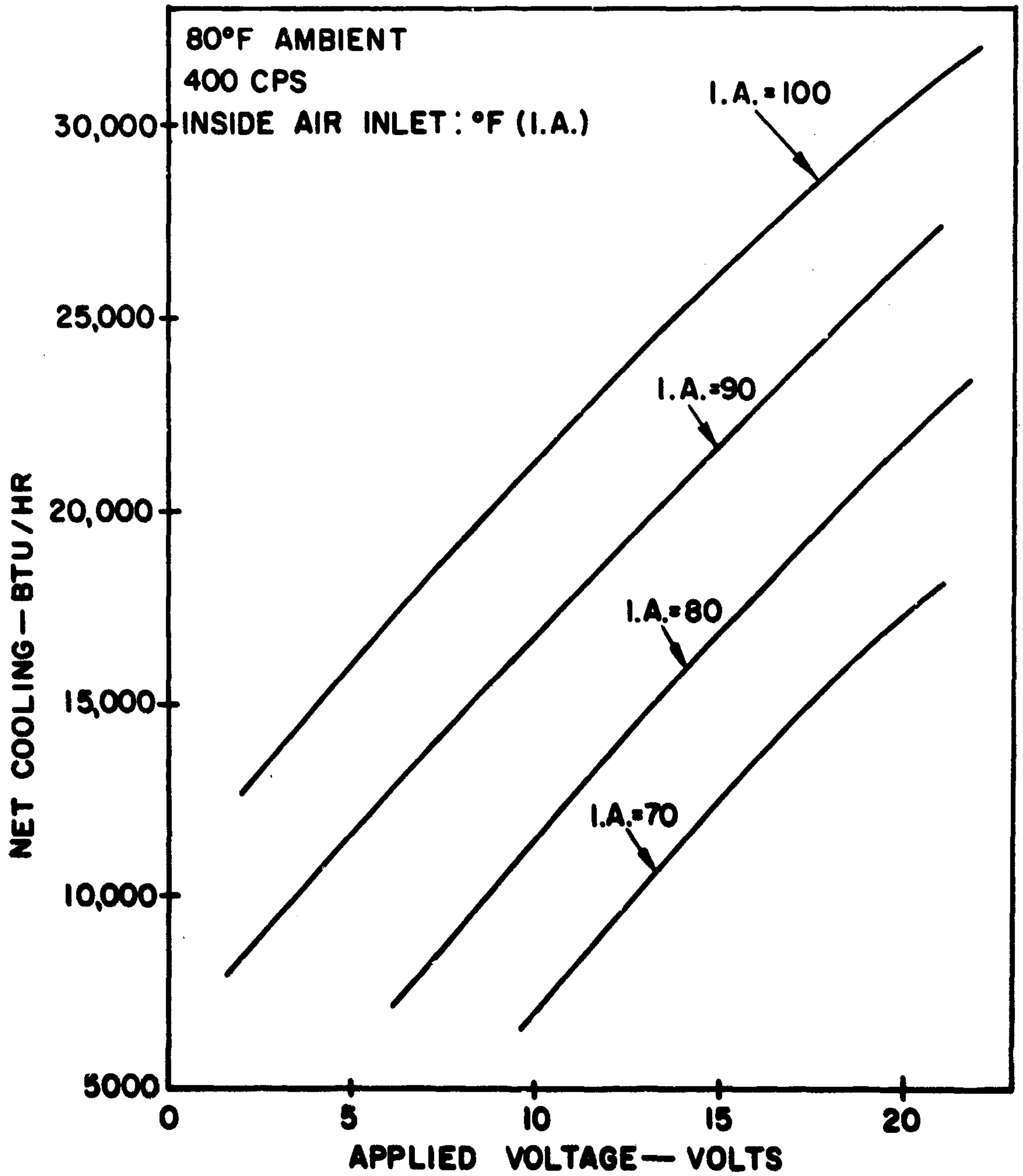


FIG. 4-4

has a total weight of 41 pounds and is fabricated from 6061-T6 aluminum and polyester plastic. Individual weight distribution of the frame, case and other components are given in Section 4.1.2.

The structure is designed to withstand 5 g impact loads and internal and external vibrations in any direction. To obtain a lightweight, high strength structure to withstand these loads, a semi-monocoque construction is employed. High reliability is achieved with this arrangement by having redundant loadings.

Construction of the case is of two pieces. This consists of a molded glass impregnated polyester plastic form covering the top, bottom, half sides and inside wall. The outside wall is attached separately for maintenance and assembly purposes. A 1/16" thick case is used to obtain the necessary rigidity and structural stability. Local stiffening is used around the inside air inlet so that external ducting may be attached. Bonding means are used to attach the major portion of the case to the frame. Supplemental support is obtain by the use of rivets.

Loads from the fans and thermoelectric cores are carried through the outside air duct wall, thermoelectric mount frame and outside wall into the mount plates. These plates, with the case and frame, can be seen in Figure 7-1 and 7-4. The power supply package load is put into the structure at the top of the unit. From here, the load is carried through the case and frame to the mount plates.

The frame is made of 6061-T6 aluminum angles and flat sections. Six horizontal flat sections on the sides of the frame are used for mounting the unit and transmitting the load onto the mounts. Joining of the frame members is accomplished by welding processes.

4.2.2 Fans

One basic design is used in various combinations to handle the outside and inside air flows. In this way, the most economical utilization of fan design is achieved. This is accomplished by having the performance curves for the two air streams match the performance curve of the fan. Figure 4-5 shows the operating points of the inside

and outside fans at the nominal cooling condition.

Two fans are used to move the outside air and on fan is required for the inside air. Figures 7-2 and 7-3 show the air flow paths and fan locations. A fan for the outside air has an operating capacity of 900 cfm and inside air flow of 840 cfm is used.

To achieve a low noise, long life, low weight and small size fan; a vane axial design has been selected. A 10,000-hour overall fan life may be expected when running on 208 volts A.C., three phase, at 3,450 or 3,700 rpm for 60 or 400 cycle operation. A fan with a dual wound motor can be incorporated in the system design. This fan would give approximately identical performance as the individually wound motors.

In trying to meet a unit noise level of ASHRAE NC 60, a relatively quiet fan is needed. Fan noise is 80 db from Figure 4-5. Incorporating this noise plus other system noises and relating this to a person standing two feet in front of the environmental control unit, gives an average sound level of 65 db.

The fan is made of aluminum alloy and each motor wound fan has the following weights:

60 cycle wound - 18.50 lbs.

400 cycle wound - 14.25 lbs.

60/4000 cycle wound - 23.25 lbs.

Overall fan efficiencies from different fan configurations and operations can be expected as follows:

60 cycle fan, 60 cycle operation	48%
400 cycle fan, 400 cycle operation	54%
60/400 cycle fan, 60 cycle operation	50%
60/400 cycle fan, 400 cycle operation	46%

Location of the inside air fan is vertically along the inside wall at the top of the unit. It is accessible from the outside because the outside air duct is split in the middle after clearing the upper thermoelectric core. A flexible rubber bellows is attached to the outside diameter of the fan from a plastic flange on the case. Six holes are located in this flange to accept external ducting to

FAN PERFORMANCE CURVES

FOR 60 CYCLE, 3 PHASE, 208 VOLT A.C.

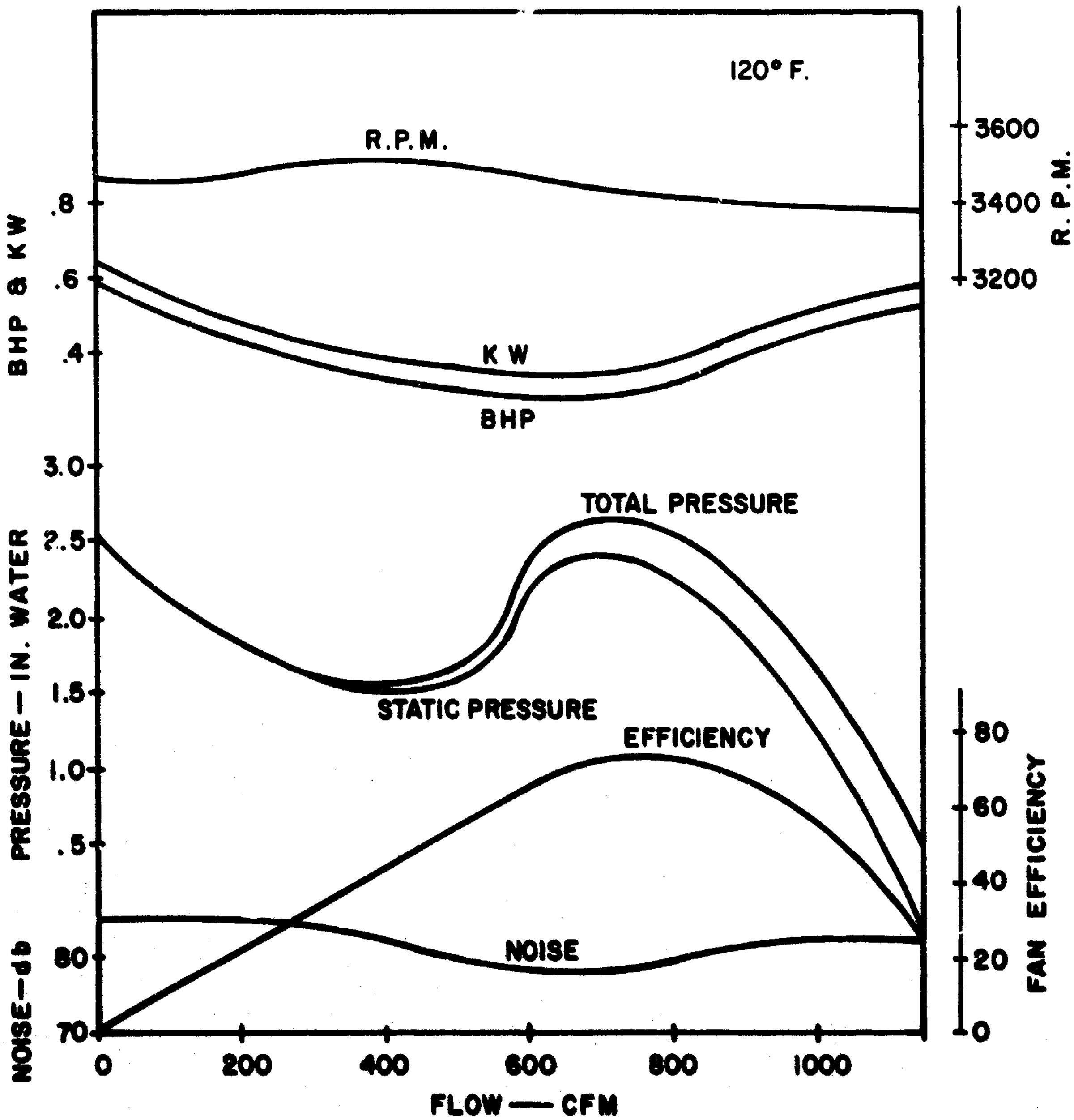


FIG. 4-5

the unit. Other bolts serve to constrain and locate the fan to the inlet duct.

The outside air fans are mounted horizontally on the outside air duct near the top and outside wall. These fans are located as close to the side walls as practical. They use the same mounting flange on the fan as used on the inside fan.

Seals are used to ensure a leak tight mating connection on both ends of the fans. A rubber joint is used to connect the outside wall to the fan. This gives an air tight assembly with flexibility to handle tolerance build-up and fan vibration.

4.2.3 Duct

By arranging the thermoelectric modules in the most advantageous locations, a minimum of internal ducting is required. The ducting is broken down into three categories, (1) outside air outlet, (2) inside air inlet, (3) inside air outlet.

Material for the outside air outlet and inside air inlet is made from glass impregnated polyester plastic. This material was chosen for the following reasons:

1. High strength
2. Low weight
3. Thermal conductivity
4. Thermal expansion
5. Heat distortion
6. Impact strength
7. Water absorption
8. Mold shrinkage

Table III gives its physical properties as well as military specifications.

Ducting for the outside air outlet serves a dual purpose, (1) carry part of the thermoelectric core and fan weight under certain loadings, (2) control the direction of air flow.

Fabrication will be of simple bend construction with no compound bent parts. This will reduce mold costs as well as obtaining the maximum conditions for certain loads. Auxiliary supports handle

other loads. The curved sections will be joined and the turning vanes will be installed by application of the duct material using a hand layup method.

TABLE III

Physical Properties of Glass Impregnated Polyester Plastic

Mold Shrinkage, In. per In.	0.002 - 0.006
Specific Gravity	1.8 - 2.3
Tensile Strength, P.S.I.	4,000 - 10,000
Compressive Strength, P.S.I.	20,000 - 26,000
Flexural Strength, P.S.I.	13,000 - 20,000
Impact Strength, 1200	1.5 - 16.0
Thermal Conductivity *	10 - 16
Thermal Expansion, 10^{-5} per °C	2.5 - 3.3
Heat Distortion Temperature, °F	7400
ARC Resistance, Sec.	120 - 180
Water Absorption, 24 hr. 1/8 in. thk., %	.06 - .28
Military Spec	MAI - 60

*10^{-4} CAL per Sq. CM, per I°C per CM

The duct is of monocoque construction with a small percentage of the loads transmitted through the assembly joints. Shunting of the static and dynamic loads are accomplished by mounting the duct in this case in such a manner as to allow the loads to be carried by the case and frame. Figure 7-1 shows the angles mounted on part of the duct which carries some of the core load into the case and frame. These angles are made of aluminum alloy material.

The inside air inlet duct is made of the same material as the outside outlet duct. Joining methods are also of the same method as used above.

Inlet chambers above the thermoelectric cores are made of glass impregnated polyester plastic, which is attached by screws to the outside wall and thermoelectric mount frame as shown in Figure 7-1. These chambers are removed with the outside wall. This will provide full access for maintenance of the thermoelectric cores.

4.2.4 Thermoelectric Cores

In the thermoelectric unit there are two thermoelectric cores. Each core consists of twelve modules for a total of twenty-four modules per unit. Locations of the cores are along the outside wall, one above the other, see Figure 7-1.

Shunting is accomplished on either end of the modules. The shunt is made from aluminum alloy and joins module to module with a 3".00 by 2".20 strap which is bolted in place. Figure 7-1 shows their locations.

Each module has an overall size of 7.500 inches by 10.375 inches by 3.050 inches. The weight of each individual module is 5.15 lb. The module arrangement is shown in Figure 7-5.

Individual modules are held in place by supports mounted horizontally on the outside air outlet duct, see Figure 7-5 and with a support located behind the module shunts. Supporting bolts are inserted from underneath the supports and modules.

Perpendicular sealing around the modules from the outside air flow is accomplished by polyurethane foam attached to the modules. Intermodule sealing is performed by plastic sheets bonded and formed to interlock between modules. A track is formed by using

the plastic sheets which guide the modules into the unit.

Current flow into the modules enters through a hot side fin base in one of the corners and exits out of the hot side fin base at the diagonally opposite corner. In order to achieve one module design, adjoining modules are inverted alternately. This also facilitates the use of a short shunt. The flow of current through one couple is as follows. Current passes through a hot side fin base and pillar, goes through a pellet, through the coil side fin base, fins and out through another cold side fin base, pellet and back into a hot side pillar and base.

All fins, pillars and fin bases are made of 1100 aluminum alloy. Geometry of the fins is of the staggered strip fin design with the following dimensions.

18 fins per inch
.125 inches deep
.010 inches thick
.471 inches high for inside air flow
.683 inches high for outside air flow

Fin bases for both flows are 2.200 by .978 inches with a thickness of .020 and .060 inches for inside air and outside air respecitvely. The pillars are .125 inches thick with .062 inch high risers. The risers are square in section with the same dimensions as the pellets.

In the extensive program on the development of thermoelectric materials at Westinghouse, the most suitable materials found for cooling applications were basically a bismuth-tellurium-antimony compound for the P-type and a bismuth-tellurium-selenium compound for the N-type. Some improvements were obtained with small amounts of additives. Studies at the Westinghouse Research and Development Center and during manufacture at the Westinghouse Youngwood Semi-conductor Division led to materials which, when assembled as couples in a "Direct Transfer" type of cooling module, had an average couple figure of merit Z of $2.62 \times 10^{-3}\ {}^{\circ}K^{-1}$ and average material constants as follows:

TABLE

Item	Units	P-Type	N-Type
	V/°C	185	195
	ohm-cm	9.5×10^{-4}	1.0×10^{-3}
k	watts/°C cm	1.075×10^{-2}	1.75×10^{-2}
Z	$\frac{1}{°K}$	3.0×10^{-3}	2.7×10^{-3}

In order to obtain good mechanical strength, the processing of the above materials was by the pressed and sintered method for the P-type and the Bridgman method for the N-type. A notable development was the anisotripic P-type pressed and sintered bismuth telluride. This material has very good physical strength and performance. This is very important in cooling applications where the resistance to thermal shock is very necessary. Bridgman P-type bismuth telluride has work cleavage planes and is subject to crumbling when loaded or thermal shocked. On the other hand the Bridgman N-type material has good physical strength.

The pellets are fabricated from material of the Bismuth-Telluride family. Each pellet is .47 inches square with a length of .062 inches.

The case material is made from glass impregnated polyester plastic. Choice of this material is for the same reasons given on the case are .060 thick with stiffing ribs on both sides of the case. In using a heavy wall with stiffeners no warpage will occur after meltdown and with the use of screws on the ends, a compressive pre-load is applied to the pellets with minor deflection of the case.

An anit-briding gasket is inserted around the pellets. These gaskets also serve the purpose of locating the pellets during assembly.

4.3 Module Fabrication

4.3.1 Module Assembly Procedure

The parts list on Figure 7-5 shows a typical thermoelectric module assembly. An assembly procedure is as follows:

1. Hot side fins, fin base and pillar and cold side fins and fin base are soldered into separate sub-assemblies at a high temperature.
2. Pellets are mounted into the plastic sheet spacers.
3. The hot side fin sub-assemblies are inserted into the module cases.
4. Fixturing these sub-assemblies are as follows:
   a. Half of a module case sub-assembly is positioned.
   b. Plastic sheets are located on the case.
   c. Cold side fin sub-assemblies are mounted.
   d. Another layer of plastic sheets are placed on assembly.
   e. The second half of the module case sub-assembly is installed.
   f. Bolts are inserted into the module case.
5. Pressure plates then compress the entire assembly for the lower temperature melt down.
6. Before cooling down and while the pressure plates are still on the assembly, the bolts are torqued to give the proper pre-load on the pellets.
7. The pressure plates are then removed and the assembly is complete.

4.4 Component Alterations

4.4.1 Reversibility

Being able to reverse the inside air flow into the thermoelectric unit is accomplished by interchanging the outlet screen to the other corner of the top side. Six screws are removed from the screen and likewise from the coverplate on the opposite side. A polyethylene seal is used on both parts to seal the air between the screen and seal. Figure 7-1 shows the location of parts.

Movement of parts is carried out from the inside of the room. A little more time consuming process is required if interchangeability must be accomplished from the outside wall of the case. This would require removal and replacement of the power supply package and inside air flow.

4.4.2 Maintenance

All repair and replacement of parts in an installed thermoelectric unit is carried out from the outside wall. Figure 7-6 shows the component removal sequence from the unit. To replace failed inside fan motor, rectifier or inoperative module the following steps are necessary:

Failed Rectifier

Step 1. Remove outside wall.

Step 2. Remove cover plate screws on power supply and capacity control package.

Step 3. Loosen electrical connections.

Step 4. Remove hold down nut of spark plug rectifier.

Step 5. Install new rectifier with hold down nut.

Step 6. Reverse steps 1, 2, and 3.

4.4.3 Failed Inside Air Fan Motor

Step 1. Remove outside wall.

Step 2. Loosen electrical connections on outside air fans.

Step 3. Remove outside air fans.

Step 4. Loosen all electrical connections to Power Conversion and Capacity Control Package.

Step 5. Remove gusset hold down bolts to power supply.

Step 6. Remove Power Conversion and Capacity Control Package.

Step 7. Loosen electrical connections to inside air fan.

Step 8. Remove hold down bolts on fan.

Step 9. Lift flexible rubber duct connection.

Step 10. Slide fan horizontally out toward outside of case.

Step 11. Remove nut and key from impeller.

Step 12. Remove impeller.

Step 13. Loosen motor screws in bell housing.

Step 14. Remove motor.

Step 15. Reverse Steps 1 thru 14.

4.4.4 Inoperable Thermoelectric Module

Should a module resistance become too large, it is felt that an automatic by-pass on the modules would become too expensive and

complex to be worthwhile. Having twenty-four separate modules, a module can be shunted manually with only a 6% reduction in performance. This is accomplished as follows:

Step 1. Remove outside wall.
Step 2. Remove filter.
Step 3. Remove shunt bolts on failed module.
Step 4. Loosen holding bolts on module.
Step 5. Remove inoperable module.
Step 6. Attach module shunt to module.
Step 7. Reverse steps 1 thru 5.

4.2.5 POWER CONVERSION AND CAPACITY CONTROL PACKAGE

The Power Conversion and Capacity Control Package contains the electrical components necessary to convert alternating current to a variable magnitude of direct current. This direct current, when applied to the thermoelectric modules, will result in a modulated heat removal (or generation) capacity which is dictated by the heat load and, in turn, controlled by a temperature sensing device.

The essential electrical components of the Power Conversion and Capacity Control Package are represented in the following block diamgram (Figure 4-6).

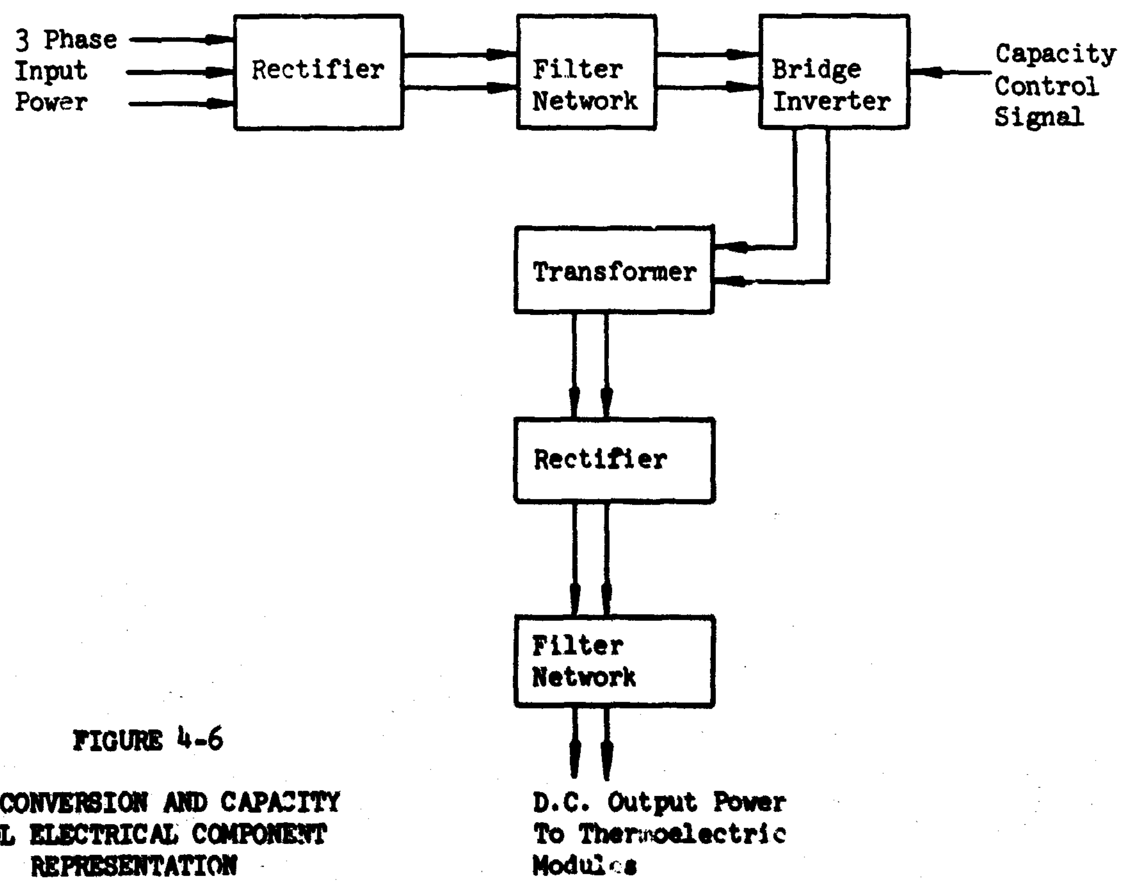


FIGURE 4-6

POWER CONVERSION AND CAPACITY CONTROL ELECTRICAL COMPONENT REPRESENTATION

The three phase input power will consist of 208 volts with a frequency of 50 cycles per second or more. This input power is initially applied to a full wave bridge rectifier and filter network. The resulting d.c. voltage is then converted to a high frequency pulse-width-modulated wave form. The time duration of the pulse width is governed by an electronic logic system which is controlled by a selector switch and a temperature sensing device. The magnitude of the square wave inverter output voltage is reduced by utilizing a step-down transformer. The weight and size of such a transformer is very small since the operating frequency is 2000 cycles per second. The transformer low voltage output is rectified, filtered and applied to the thermoelectric modules. This power conversion and capacity control design provides continuous control from full to minimum output voltage and maintains the ripple distortion below 10% at all voltage levels.

Power Conversion System Design Considerations

The power supply will operate with an input of 50 cps or above. The line voltage is rectified by a full wave bridge producing approximately 280 volts dc (refer to Figure 6-1 for schematic diagram). The bridge circuit results in a relatively high dc voltage, and use twice as many rectifiers as the half wave circuit, but was chosen for the following reasons:

1. No half wave or dc component of current in the input wave form.
2. In half wave operation the current would be twice as high and the neutral lead would carry full dc current which would exceed 25 amps.
3. The rectifier losses in the two circuits are equal so there is no loss in efficiency.
4. The ripple frequency of the bridge circuit is twice as high as the half wave circuit, and results in about one fourth the ripple voltage. Both of these factors make filtering of the dc component easier which results in a smaller, lighter and less expensive filter.

The filtered dc is then applied to an SCR bridge inverter utilizing a step down transformer. The bridge inverter was chosen over a half wave circuit because the half wave circuit results in twice the peak inverse voltage on the SCR's. The higher PIV would mean that two SCR's would have to be connected in series, resulting in the use of the same number of SCR's and the same losses. The bridge circuit results in lower power handling requirements for the transformer primary winding which results in a ligher transformer.

The output of the transformer is full wave rectified utilizing a center tapped winding. The center tapped winding results in a slightly larger transformer, but requires only one-half the rectifiers and results in only one-half the losses when compared to since the dc output voltage is low and the current is high. The dc output is then filtered to maintain the ripple below the desired value. The SCR's, because of an inherent higher forward voltage drop, are located in the high voltage-low current portion of the circuit and the rectifiers are in the low voltage-high current portion of the circuit to keep losses to a minimum. The minimum efficiency of the power conversion unit shall be 87% at full load. The efficiency is independent of input frequencies, above 50 cps.

The components and materials to be used are of high quality and have high temperature capabilities to keep the reliability high and the weight and size low. All rectifiers, SCR's and transistors are of the silicon type, and all transformer materials will be high performance, high temperature materials.

Power supply cooling will be provided by outside ambient air at a static pressure of 1.4 inches of water whenever the outside fans are operating. During modes of operation when the outside fans are not operating, cooling will be provided by the inside fan and airflow will be controlled by a solenoid operated valve. This valve will be actuated by the polarity transfer breaker.

The power conversion unit and portions of the capacity control system will be packaged in a metal cabinet with maximum dimensions of 20" x 8" x 7.75". Printed circuit boards that contain portions of the capacity control system will be mounted on the back side of the control panel.

A general assembly drawing of the proposed power conversion and control package is illustrated in Figure 7-8.

4.2.6 CONTROL AND MONITOR PANEL

The Control and Monitor Panel contains the components necessary for system operation selection, malfunction indication, and protection. The basic components associated with the control and monitor panel are indicated in the block diagram of Figure 4-7 and the diagramatic representation of Figure 4-8. The general assembly drawing of the control and monitor panel is shown in Figure 7-9.

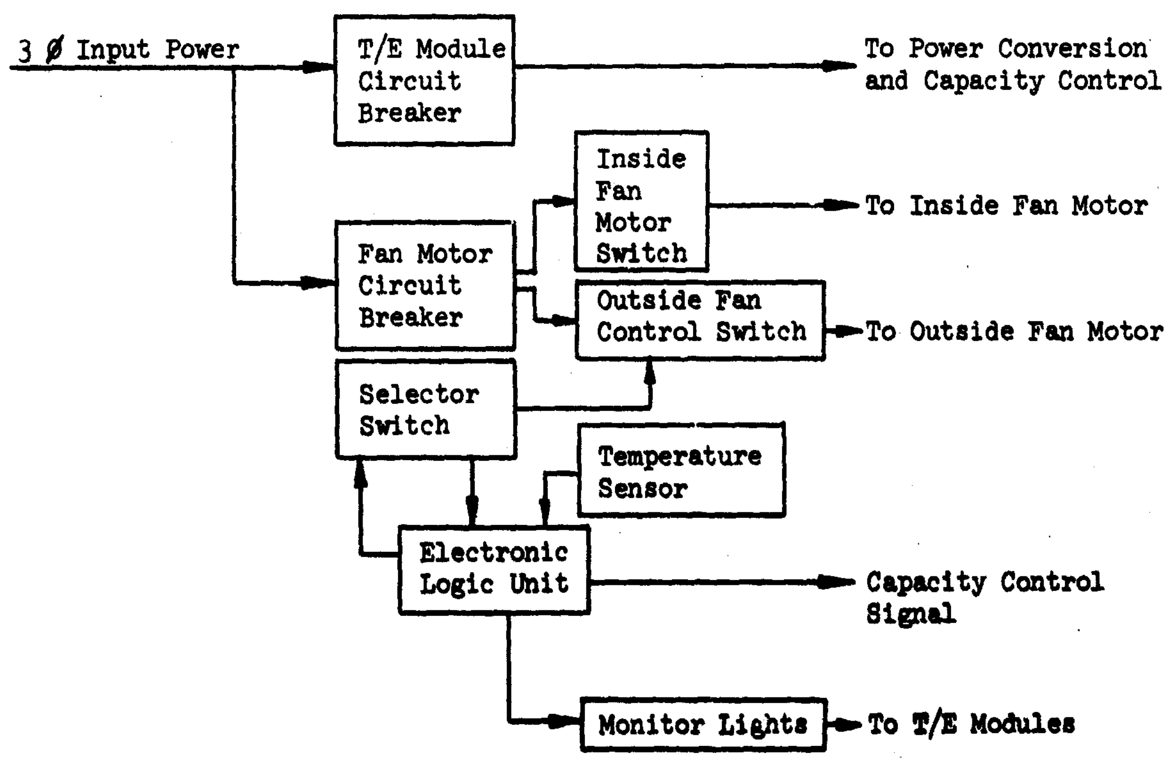
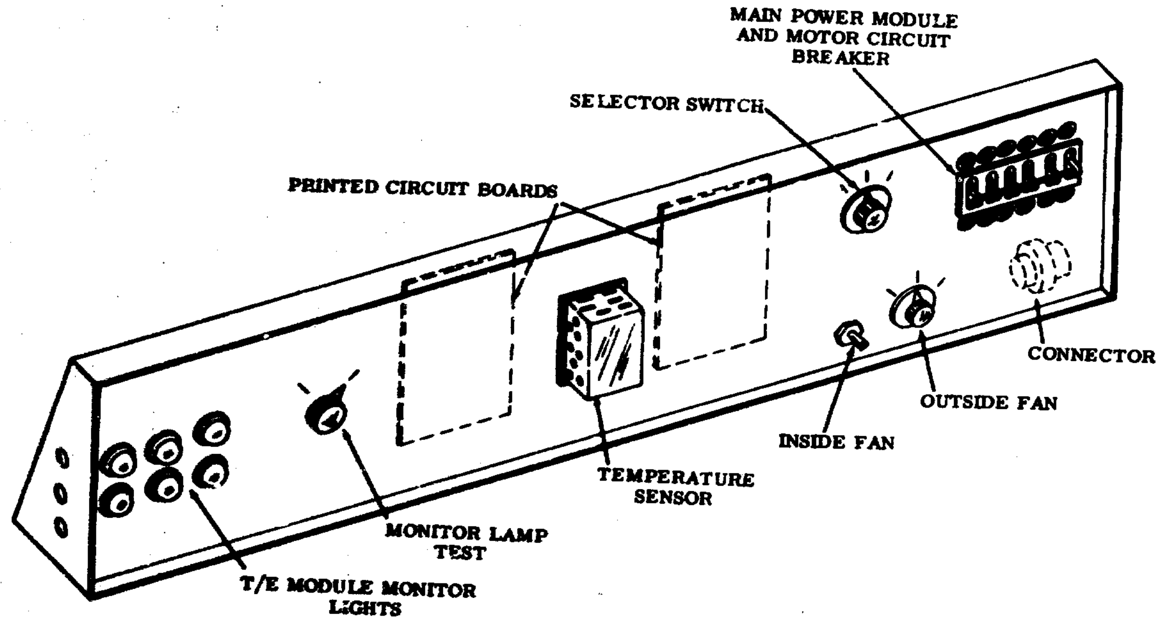


FIGURE 4-7

BLOCK DIAGRAM OF THE CONTROL AND MONITOR PANEL



CONTROL AND MONITOR PANEL
FIG. 4-8

4.2.6.1 Main Power Circuit Breakers

Two Heinemann circuit breakers, Type AM 333, will connect the input power to the power conversion unit and the fan motors. These circuit breakers will be mechanically connected to insure that the fan breaker is energized when power is applied to the power conversion unit. This mechanical connection, however, will permit independent operation of the fan motor circuit breaker.

4.2.6.2 Fan Motor Switches

Two fan motor switches will be mounted on the panel to control the inside and outside fans.

Outside Fan Motor Switch

This three position switch will provide control of the outside fans. The positions will be OFF, ON, and AUTOMATIC. In the automatic operating mode, power will be interrupted to the outside fans when the polarity transfer from cooling to heating occurs.

Inside Fan Motor Switch

This two position switch will provide control of the inside fan. The two modes of control are ON and OFF.

4.2.6.3 Selector Switch

The selector switch shall have three positions, Manual Heat, Manual Cool, and Automatic.

Manual Heat

With the selector switch in this position, full power will be supplied to the T/E modules in the heating mode and the automatic capacity control will be by-passed.

Manual Cool

With the selector switch in this position, full power will be supplied to the T/E modules in the cooling mode and the automatic capacity control will be by-passed.

Automatic

Connecting the switch in this position will cause the environmental control unit to perform in the following manner:

When the temperature sensor measures a temperature greater than, or equal to 78°F full voltage will be applied to the thermoelectric modules with appropriate polarity to produce maximum cooling.

The applied voltage, with polarity for cooling, will decrease in a direct proportion to decreasing temperature within the sensor temperature range of from 78°F to 72°F.

At a sensor temperature of 72°F the applied voltage will reach a minimum of 0 volts and the thermoelectric unit will act as a heat exchanger.

Within the sensor temperature range of from 72°F to 60°F all power to the thermoelectric modules and outside fans will be discontinued. Within this temperature range, at approximately 66°F, the voltage polarity, with respect to the thermoelectric modules, will be reversed in order to prepare the system for the heating mode.

When the temperature sensor measures 60°F the thermoelectric modules will receive minimum voltage with polarity required for heating.

The applied voltage to the thermoelectric modules will increase in inverse proportion to the sensor temperature as the sensor temperature varies from 60°F to 54°F.

At a sensor temperature of 54°F or less the thermoelectric unit will receive the maximum applied voltage with the polarity required to produce maximum heating.

4.2.6.4 Monitor Lights

Six 5 volt clear bulb lamps shall be mounted on the control panel. Each lamp shall be connected in parallel with a group of four T/E modules. The intensity of each lamp shall indicate the level of power applied across each group of modules.

A two position switch shall be connected as indicated in Figure 6-5 to provide a quick test for burned out bulbs.

4.2.6.5 Temperature Sensor

The temperature sensor shall consist of thermistor elements. This protected sensor shall sense the compartment temperature and dictate the mode and capacity of system operation through the capacity control system.

4.2.6.6 Connectors

There will be six connectors mounted on the control panel. These six components are:

1. Main Power Connector
2. Power Conversion Connector
3. Inside Fan Motor Connector
4. Outside Fan Motors Connector
5. Capacity Control Connector
6. Monitor Light Indication Connector

These connectors are necessary in order to remove and remotely locate the control and monitor panel.

4.2.6.7 Electronics Logic Unit

The electrical components which comprise the electronic logic unit will be mounted on two small lightweight printed circuit boards. This unit will send a signal to the power conversion components which will cause the thermoelectric modules to perform in a manner dictated by the position of the sector switch and the temperature sensor.

4.2.7 Power Conversion Unit Control and Protection

The high frequency square wave output of the bridge inverter is generated by proper sequential firing of the SCR's A, B, C and D (Refer to Figure 6-1). These components are, in turn, driven by square wave forms derived from pulse driven flip-flops with transformer outputs to provide the necessary isolation (Refer to Figure 6-2). This square wave SCR drive is used to obtain better turn-on characteristics and also to automatically provide reverse biasing during the OFF period. One flip-flop is slaved to the other to provide the proper phasing depending upon the selected mode of operation.

During manual operation the phasing is such that SCR's A and D, of the bridge inverter (Refer to Figure 6-1), turn on together and the B and C turn on together providing a maximum output which is the rectified, filtered and applied to the thermoelectric modules.

In automatic operation, the phasing is reversed such that, SCR's A and C are turned on together and B and D are turned on together to provide essentially a zero output in the absence of trigger signals. Stepless variation of power levels is obtained by sequential firing of the proper SCR's, which produces a pulse-width-modulated, bridge inverted output wave form.

The SCR driver flip-flops (Refer to Figure 6-2) and the pulse width modulator (Refer to Figure 6-3) govern the pulse width characteristic of the bridge inverter output wave form. The flip-flops and the pulse-width-modulator are triggered by a unijunction transistor oscillator which provides high energy trigger pulses and a saw-tooth modulating wave at twice inverter frequency. (Refer to Figure 6-3). This method provides exact half wave symmetry and avoids any saturation of the inverter main transformer.

The temperature sensor (refer to Figure 6-3) controls the pulse-width modulator and initiates the polarity cross-over detector. This component, therefore, controls the magnitude, and polarity (with respect to the thermoelectric modules) of the power conversion voltage

output as a function of temperature. The temperature sensor consists of a bridge network with positive-temperature-cofficient resistors in diagonally opposite arms. A third section with stable resistors is used as a reference. All outputs of the sensor are by means of differential amplifiers. This type of sensor was developed by Westinghouse for the AVE-9 Window Temperature Control Unit. This unit is incorporated on the F-28 Aircraft.

The pulse-width-modulator (refer to Figure 6-3) is a three-legged differential amplifier with one leg sensing the reference voltage and the two other legs sensing the temperature dependent voltages. One of these voltages increases with increasing temperature, while the other increases with decreasing temperature. The modulating saw-tooth voltage is applied to each temperature sensing leg. The output of each temperature sensitive leg is obtained by means of complementary symmetry connected circuits. The output of each leg is combined into one output which is differentiated to provide a trigger pulse to one of the SCR drivers.

The polarity crossover detector (refer to Figure 6-3) utilizes a conventional differential amplifier with complementary symmetry signal output to drive a relay amplifier. The relay is a hermetically-sealed crystal-can size device. A small hysteresis is built into the differential amplifier to prevent cycling of the crossover breaker at the crossover temperature.

The circuit above elements are combined are combined in such a manner so as to reduce the phase delay between successive firing or triggering of the proper SCR's as more power is needed by the T/E modules to increase the amount of cooling or heating. Conversely, as less power is required to operate the thermoelectric environmental control unit the phase delay between the successive firing or triggering of the SCR's is increased. The overall result is continuous proportional control with near unity (.95) power factor seen by the supply voltage at all times and a relatively smooth dc voltage seen by the T/E modules at all times. The

power supply, will work equally well on input frequencies of 50 cps or more as the unit contains adequate input filtering for 50 cps operation.

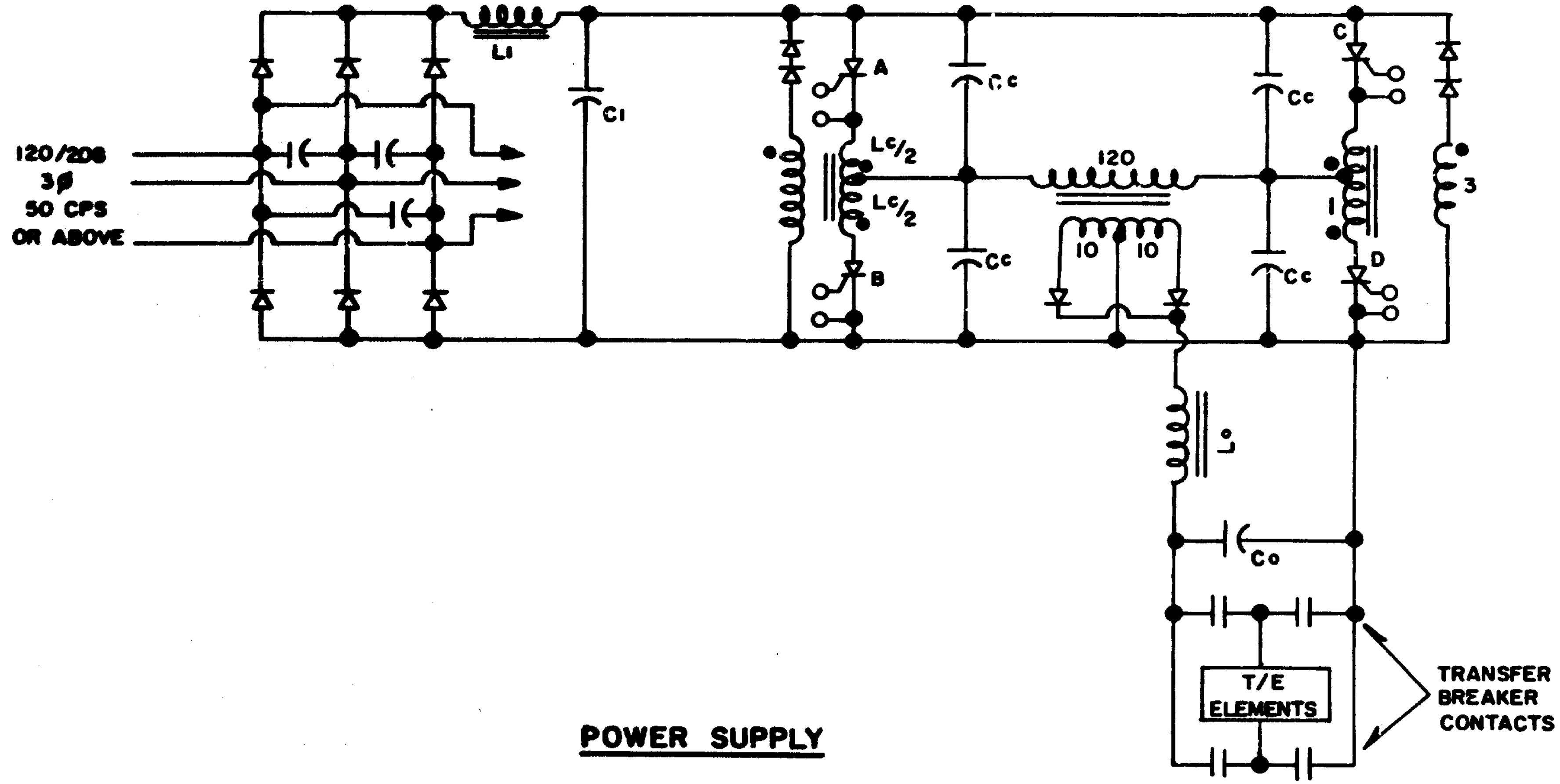
In addition to the input power circuit breakers, the thermoelectric environmental control unit is protected by an electro-mechanical safety interlock system described in Figure 6-4. The interlock mechanism will perform the following functions:

Power to the T/E modules shall be interrupted if the inside and outside fnas are turned off while the system is operating in the manual or automatic cooling modes.

Power to the T/E modules shall be interrupted if the inside fan is turned off during the manual or automatic heating mode.

5.0 BILL OF MATERIAL

P/N	QUANTITY	DESCRIPTION	MATERIAL
618J633	1	Thermoelectric Environmental Control Unit Assembly	--
773D227	1	Frame Assembly	--
773D226	24	Thermoelectric Module Assembly	--
	3	Fan-3 Phase, 60 Cycle, 208 V.AC.	--
773D243	1	Power Conversion & Capacity Control Assembly	--
	2	Brace	QQ-A-327-1
	1	Outside Wall	MIL-MAI-60
	1	Wall, Upper Chamber	MIL-MAI-60
	2	Drain Plate	MIL-MAI-60
	2	Module Support	MIL-MAI-60
	1	Case	MIL-MAI-60
	1	Inlet Duct	MIL-MAI-60
	1	Inlet Duct	MIL-MAI-60
	1	Duct Bend	MIL-MAI-60
	1	Joint	MIL-MAI-60
	1	Inlet Duct Jumper	MIL-MAI-60
	1	Outlet Duct Jumper	MIL-MAI-60
	2	Outside Air Outlet Duct	MIL-MAI-60
	1	Turning Vanes	MIL-MAI-60
	4	Angle	MIL-MAI-60
773D249	1	Control & Monitor Panel Assembly	--
	22	Thermoelectric Shunt	QQ-A-561
	1 set	Thermoelectric Leads	--
	2	Grill Assembly	--
	2	Filter	--
	As Req'd	Insulation	MIL-P-21929d
	3	Fan Seal	AMS 3208 E
	1	Duct Mount Flange	QQ-A-327
	1	Flange Stiffener	QQ-A-327
	1 set	Hold Down Hardware	QQ-A-268a
	3	Fan Mounting Flange	QQ-A-327b



POWER SUPPLY

FIG. 6-1

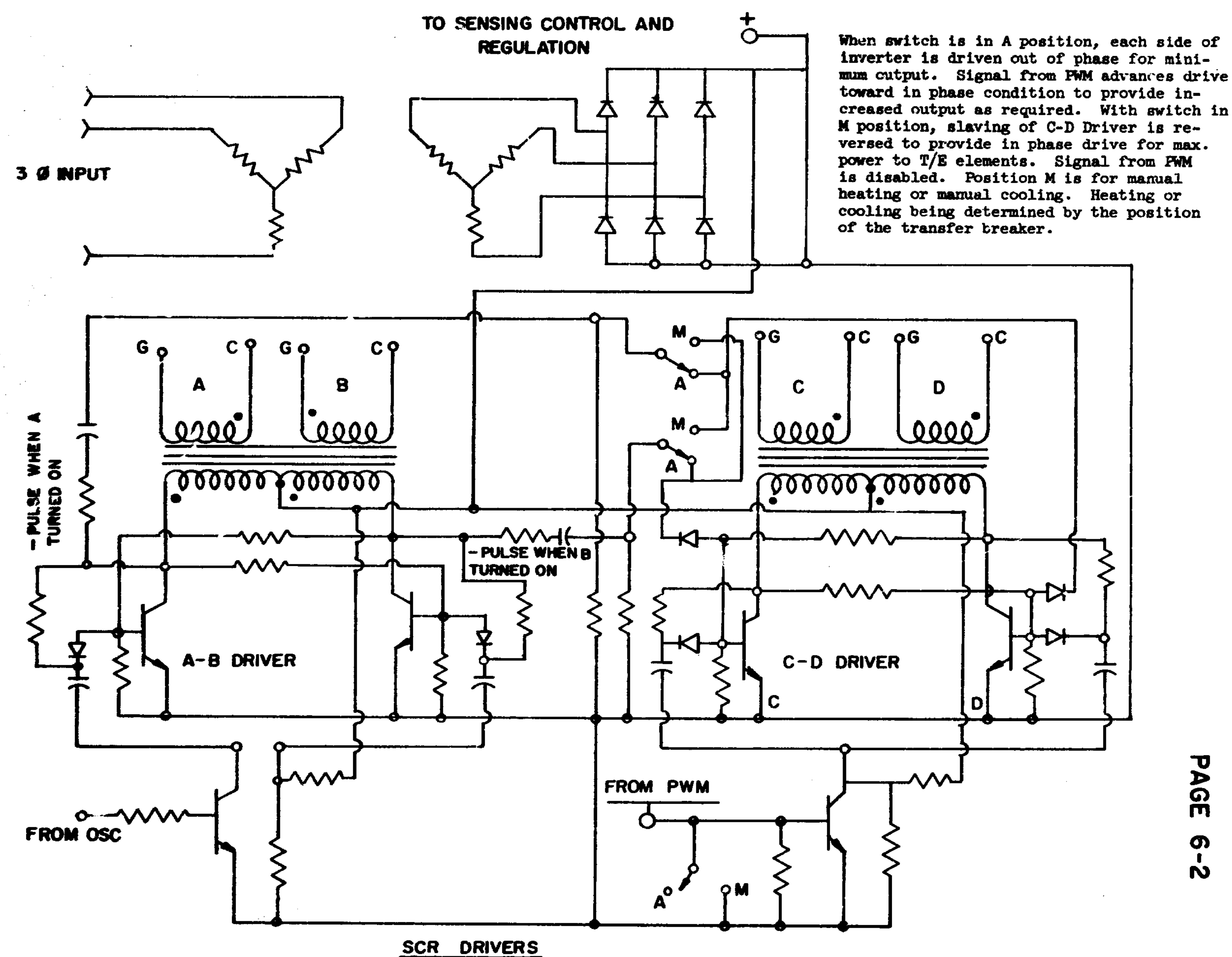
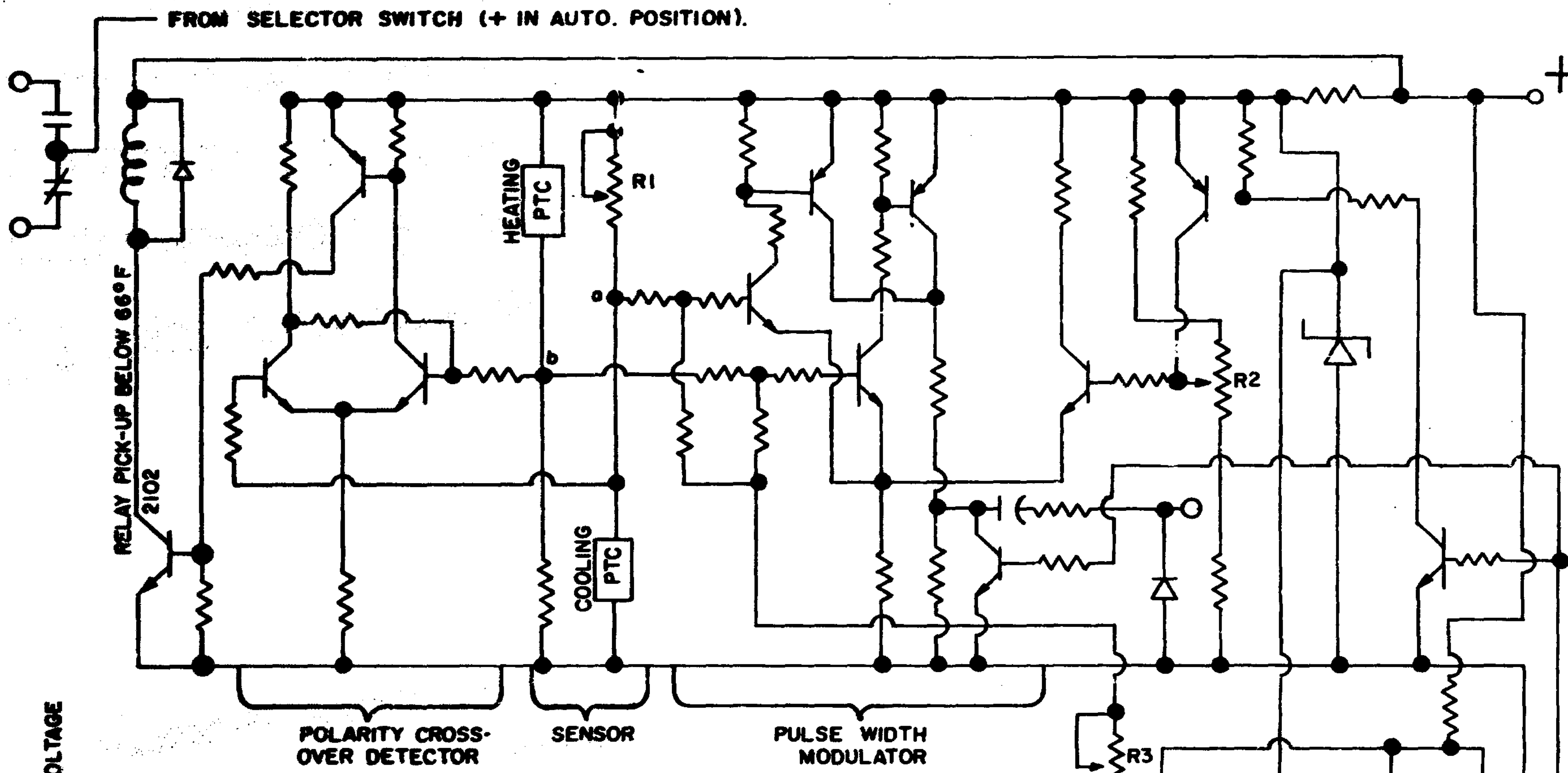
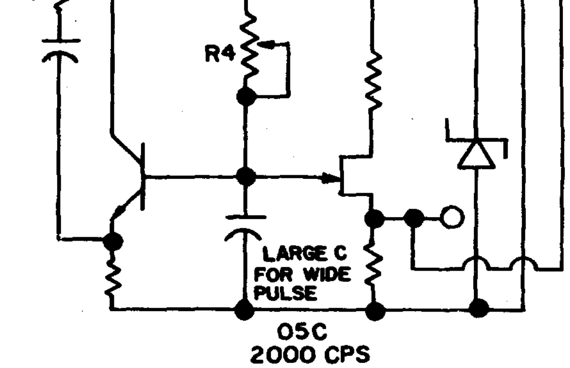
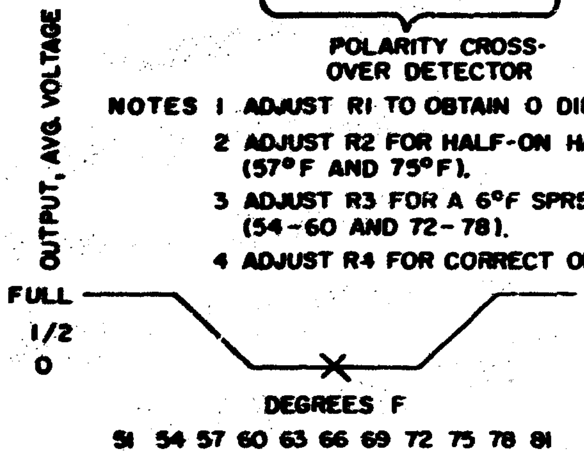

TO SENSING CONTROL AND
REGULATION
3 Ø INPUT
When switch is in A position, each side of inverter is driven out of phase for minimum output. Signal from PWM advances drive toward in phase condition to provide increased output as required. With switch in M position, slaving of C-D Driver is reversed to provide in phase drive for max. power to T/E elements. Signal from PWM is disabled. Position M is for manual heating or manual cooling. Heating or cooling being determined by the position of the transfer breaker.
G
C
G
C
A
B
C
D
M
A
M
A
- PULSE WHEN A TURNED ON
- PULSE WHEN B TURNED ON
A-B DRIVER
C-D DRIVER
C
D
FROM OSC
FROM PWM
A
M
SCR DRIVERS

FIG. 6-2



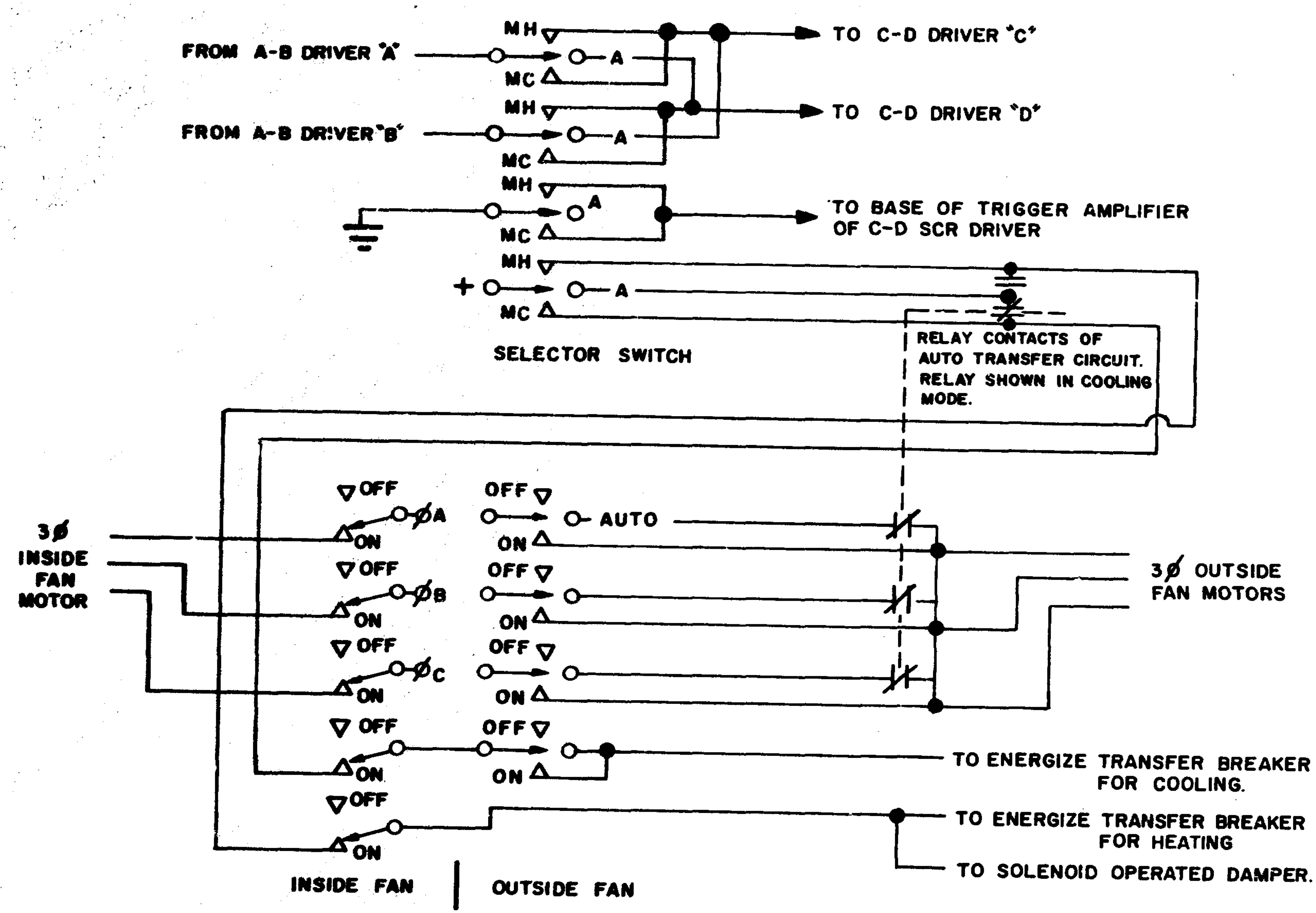


NOTES 1 ADJUST R1 TO OBTAIN O DIFFERENCE BETWEEN a AND b AT 66° F.
2 ADJUST R2 FOR HALF-ON HALF-OFF OPERATION AT 66° F ± 9° F (57° F AND 75° F).
3 ADJUST R3 FOR A 6°F SPREAD BETWEEN FULL OFF AND FULL ON (54-60 AND 72-78).
4 ADJUST R4 FOR CORRECT OPERATING FREQ. (2000 CPS).



SENSING, CONTROL AND REGULATION

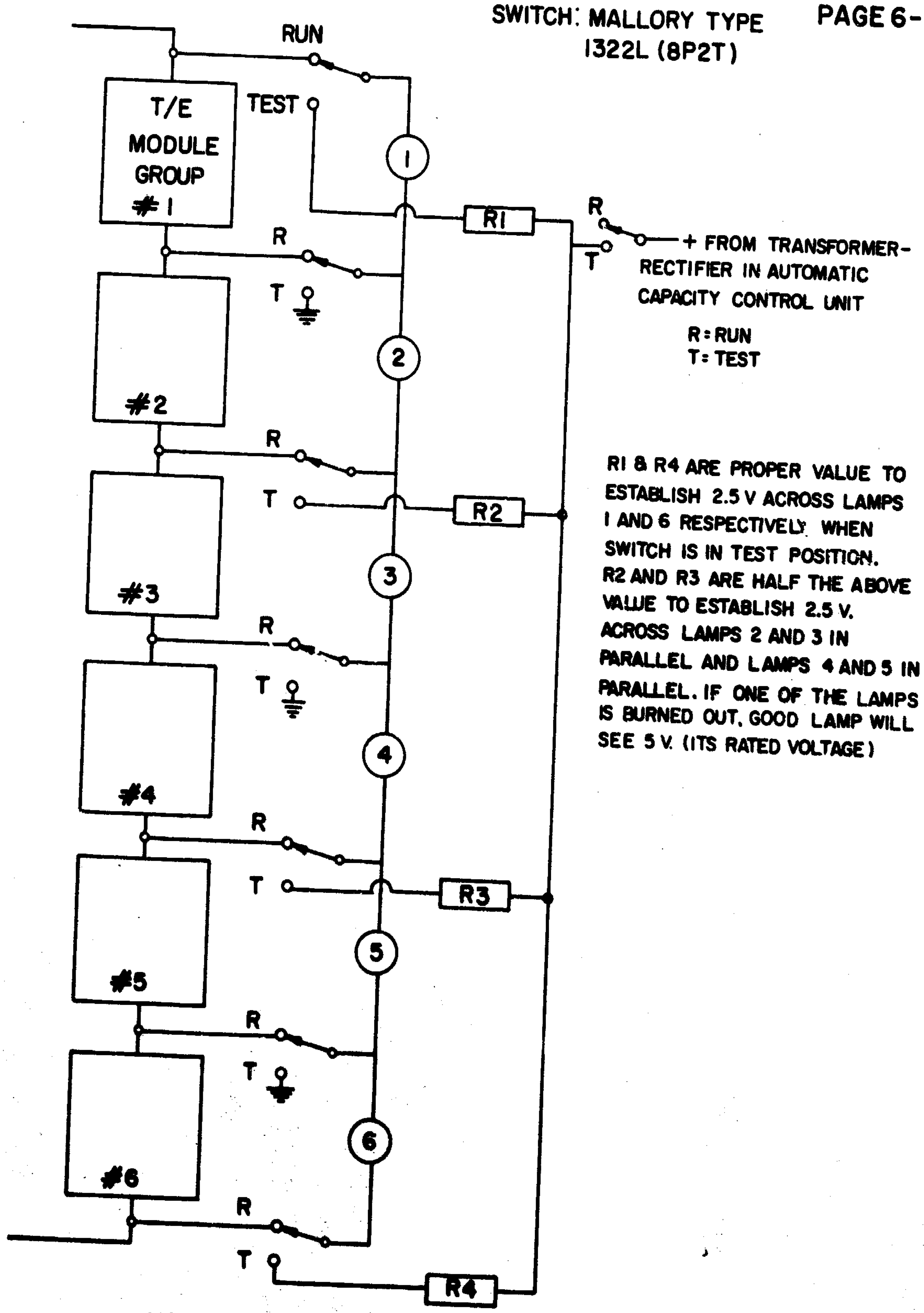
FIG 6-3



SYSTEM PROTECTION

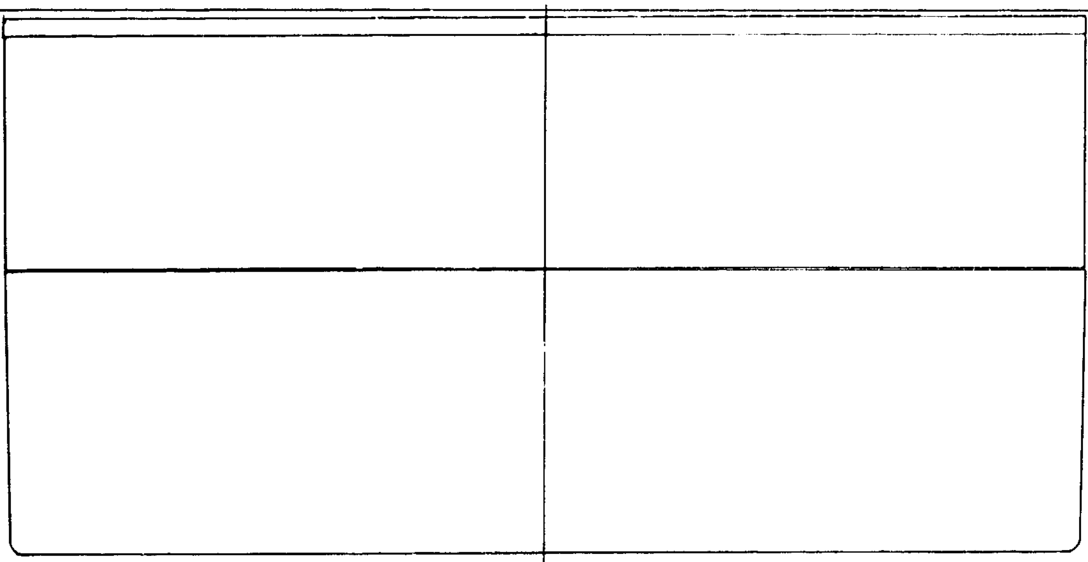
FIG. 6-4

SWITCH: MALLORY TYPE
1322L (8P2T)

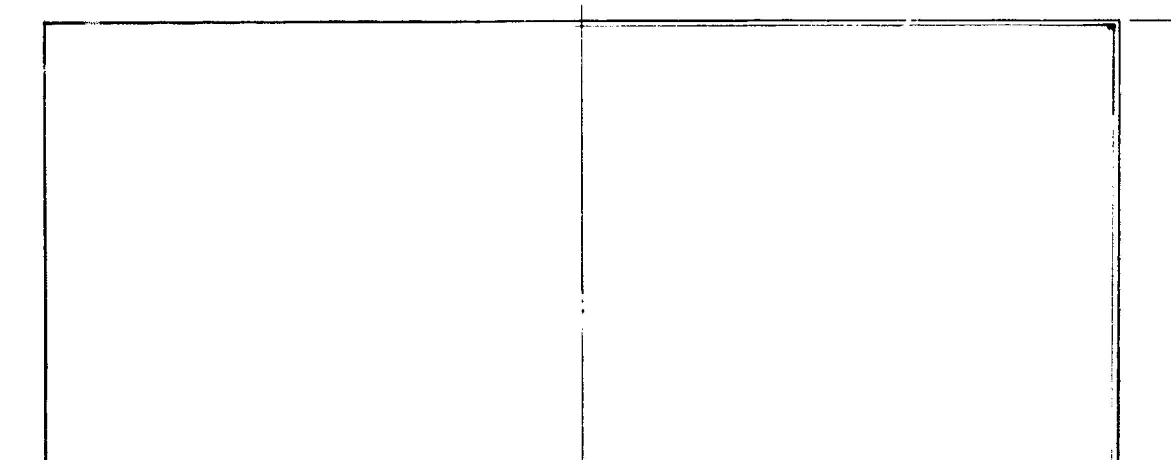


R1 & R4 ARE PROPER VALUE TO ESTABLISH 2.5 V ACROSS LAMPS 1 AND 6 RESPECTIVELY WHEN SWITCH IS IN TEST POSITION. R2 AND R3 ARE HALF THE ABOVE VALUE TO ESTABLISH 2.5 V. ACROSS LAMPS 2 AND 3 IN PARALLEL AND LAMPS 4 AND 5 IN PARALLEL. IF ONE OF THE LAMPS IS BURNED OUT, GOOD LAMP WILL SEE 5 V. (ITS RATED VOLTAGE)

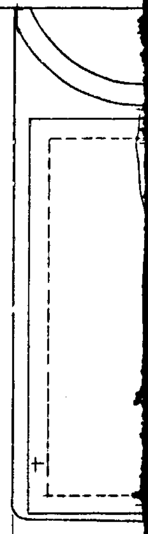
MONITOR LIGHT SCHEMATIC
FIG. 6-5



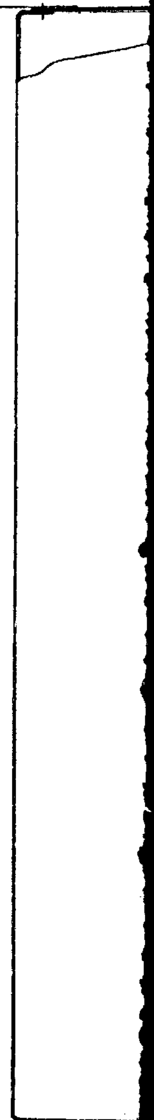
BOTTOM VIEW D-D

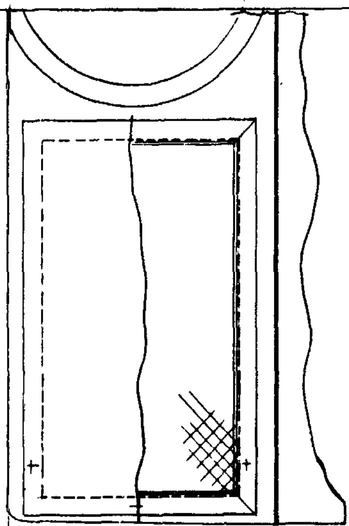


REAR VIEW

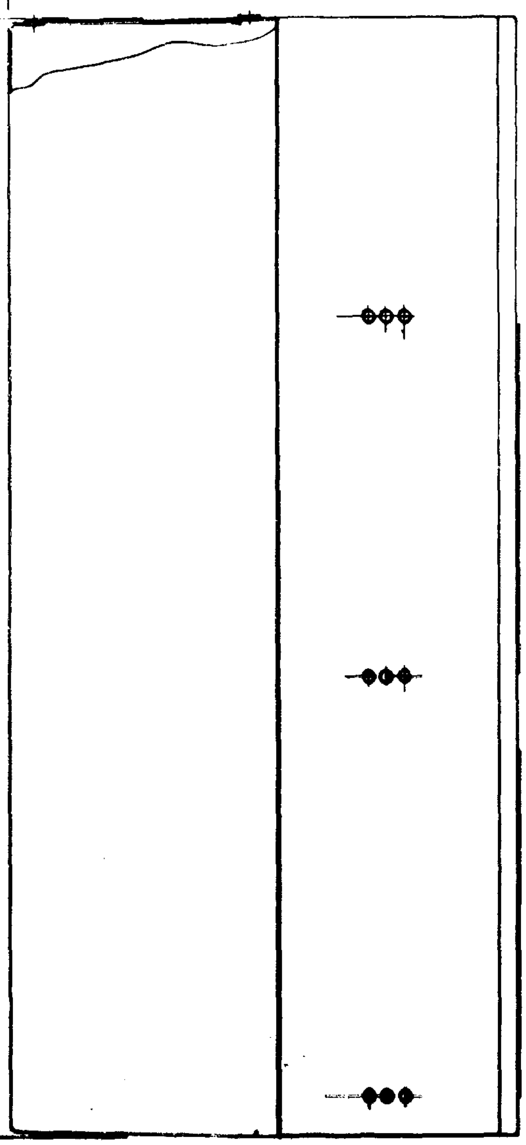


PARTIAL
SHOWING C
EXIT OPENIN
COVER OR S

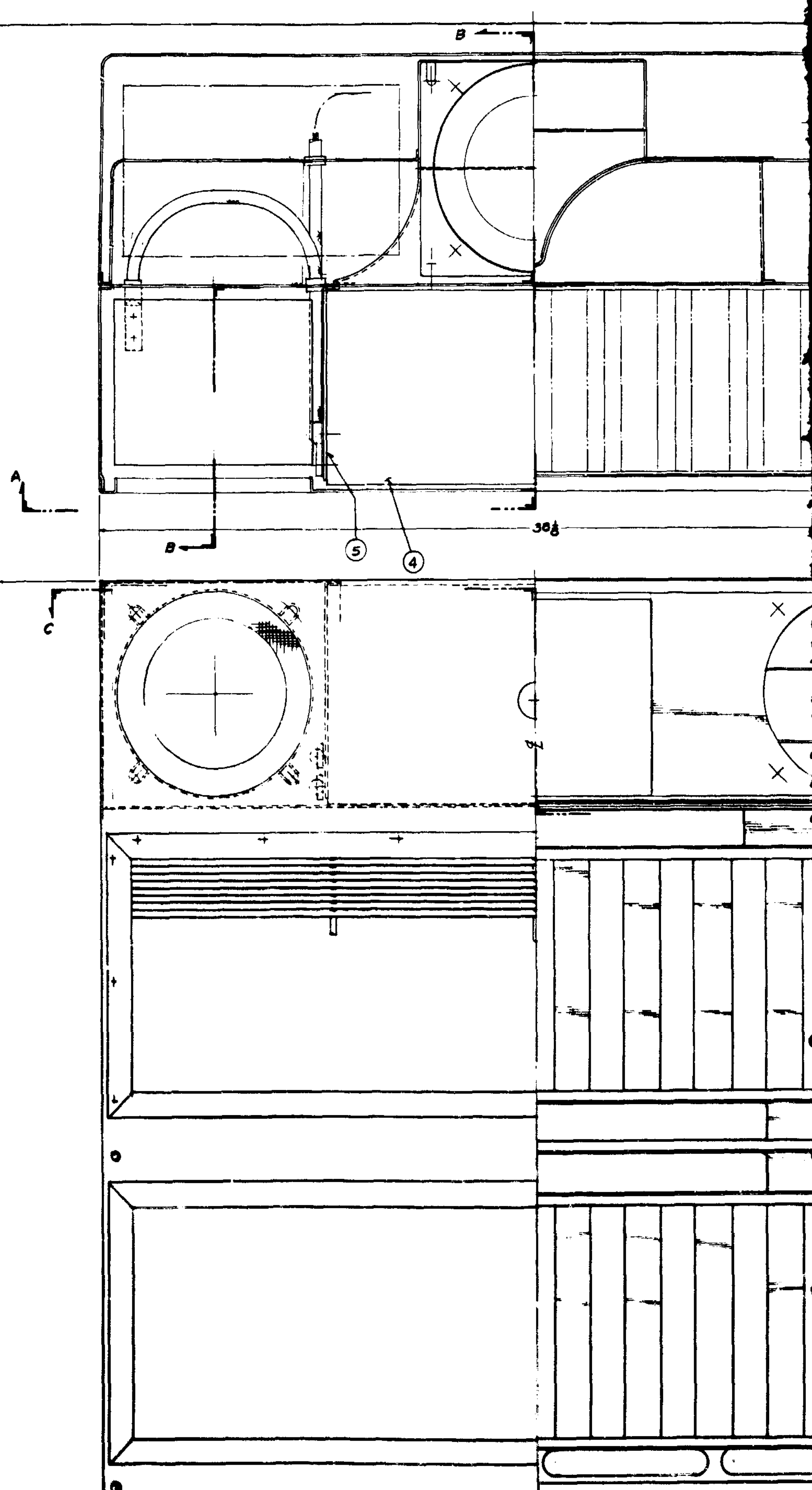




PARTIAL PLAN VIEW
SHOWING COLD AIR EXIT OPENING WITH COVER OR SCREEN



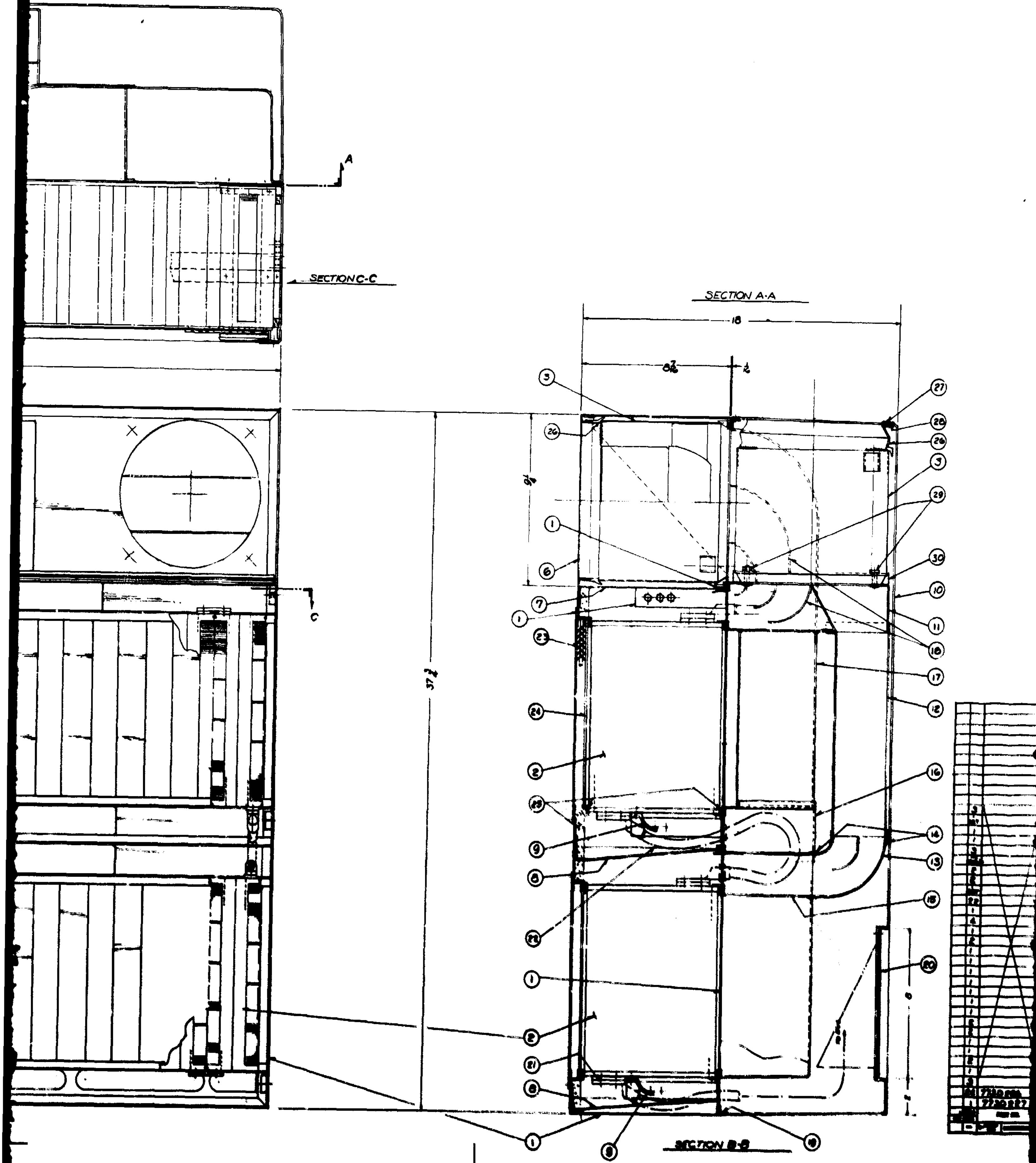
SIDE ELEVATION



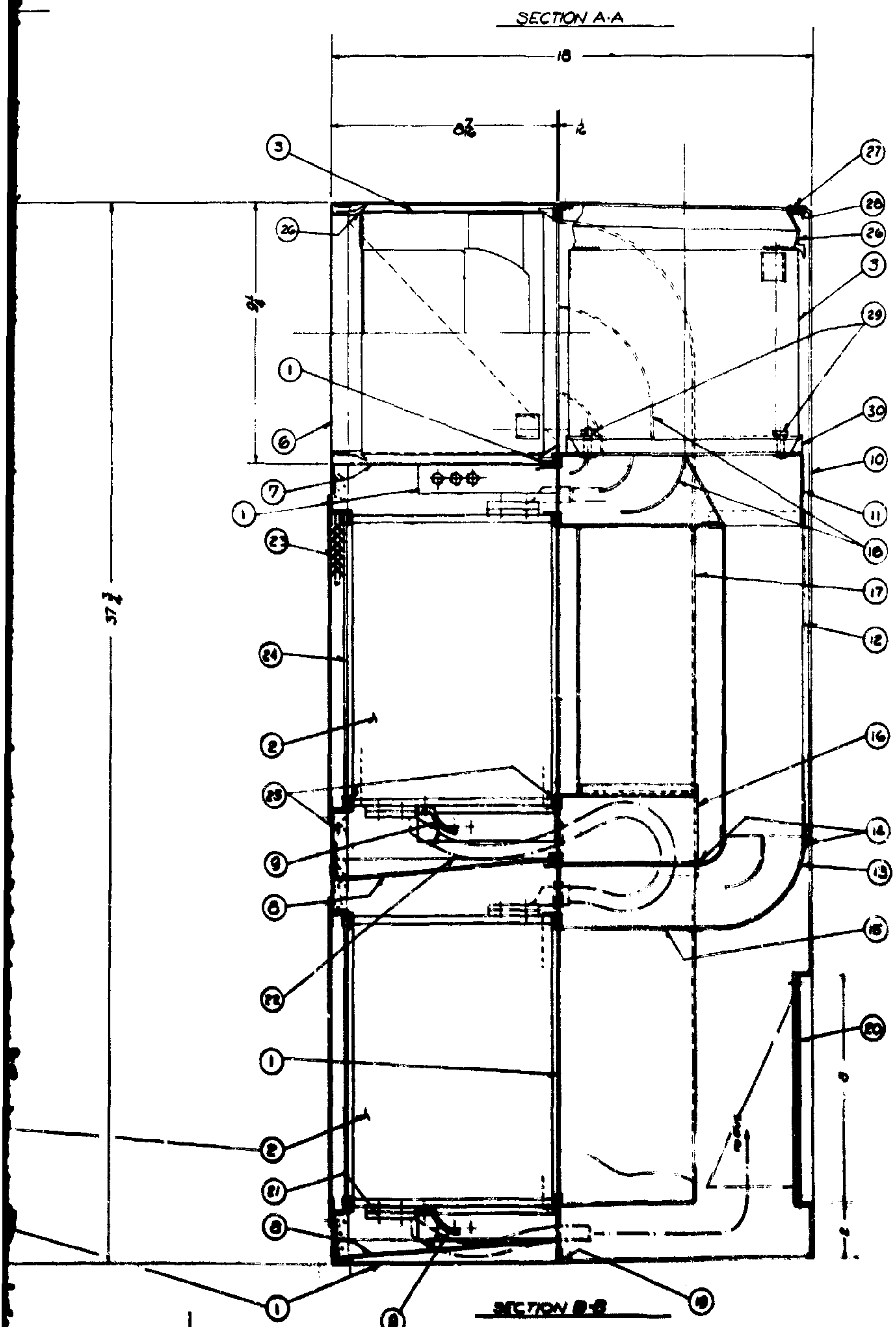
PARTIAL SECTION A-A

618J633

2



G18JG33

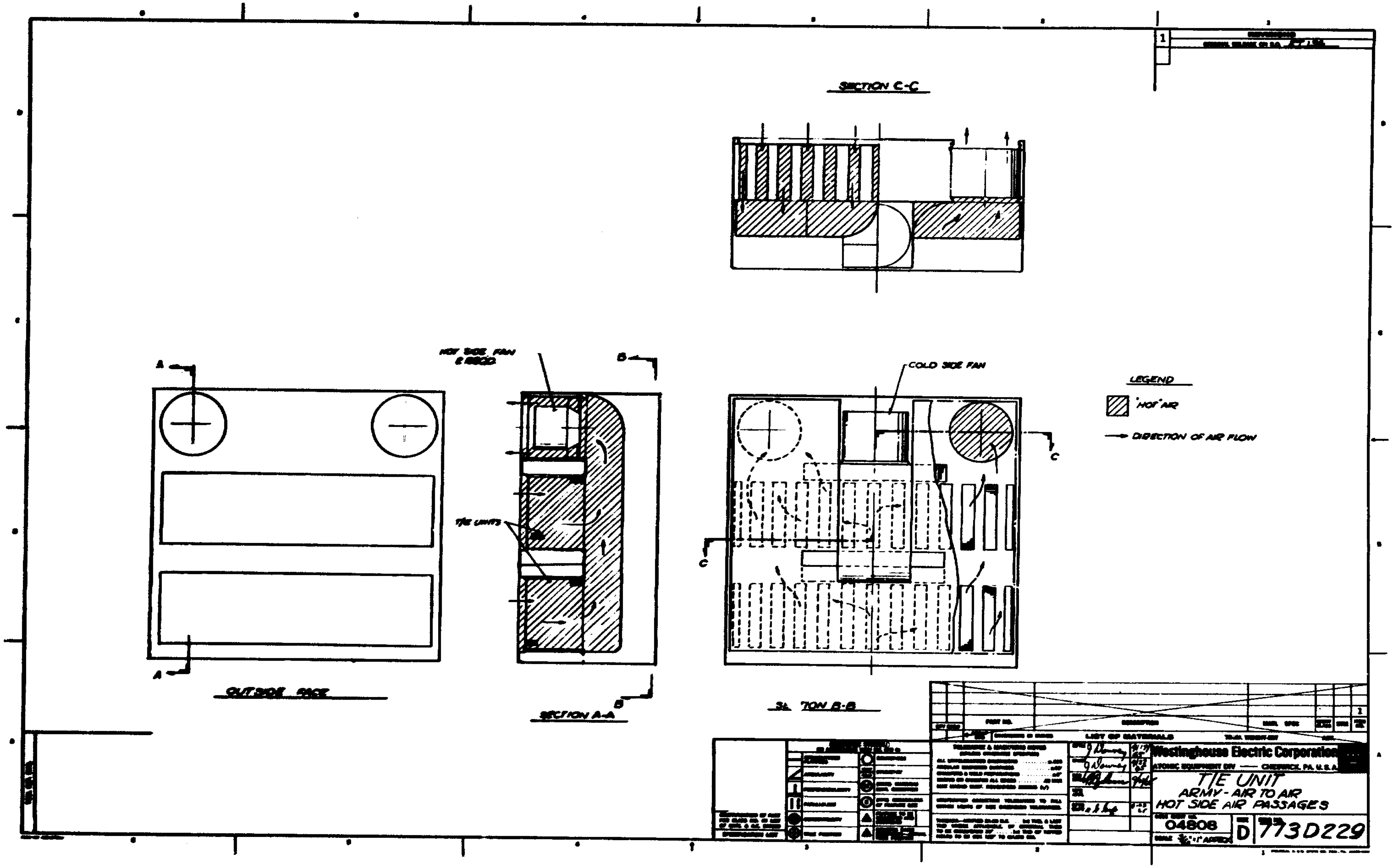


3		FAN MOUNTING FLANGE	AL	QQ-A-327	30
[illegible]		HOLD DOWN HARDWARE	AL	QQ-A-[illegible]	29
1		FLANGE STIFFENER	AL	QQ-A-327	28
1		DUCT MOUNT FLANGE	AL	QQ-A-327	27
3		FAN SEAL	NEOPRENE	AMS-[illegible]	26
[illegible]		INSULATION	POLYURETHANE FOAM	[illegible]	25
2		FILTER		—	24
2		GRILLE ASSY	AL	QQ-A-327-1	23
[illegible]		T/E LEADS		—	22
22		T/E SHUNT	AL	QQ-A-[illegible]	21
1		CONTROL & MONITOR PANEL ASSY			20
4		ANGLE			19
1		TURNING VANES			18
2		OUTSIDE AIR OUTLET DUCT			17
1		OUTLET DUCT JUMPER			16
1		INLET DUCT JUMPER			15
1		JOINT			14
1		DUCT BEND	GLASS-RESIN [illegible] MAT'L		13
1		INLET DUCT			12
1		INLET DUCT			11
1		CASE			10
2		MODULE SUPPORT			9
2		DRAIN PLATE			8
1		WALL UPPER CHAMBER			7
1		OUTSIDE WALL			6
2		BRACE	AL	QQ-A-327-1	5
1		POWER CONVERSION & CAPACITY CONTROL ASSY		—	4
3		FAN - [illegible]		—	3
[illegible]	7720896	T/E UNIT		—	2
1	7720887	FRAME		—	1

T/E UNIT
ARMY AIR TO AIR
PRELIMINARY GENL. ASSEMBLY

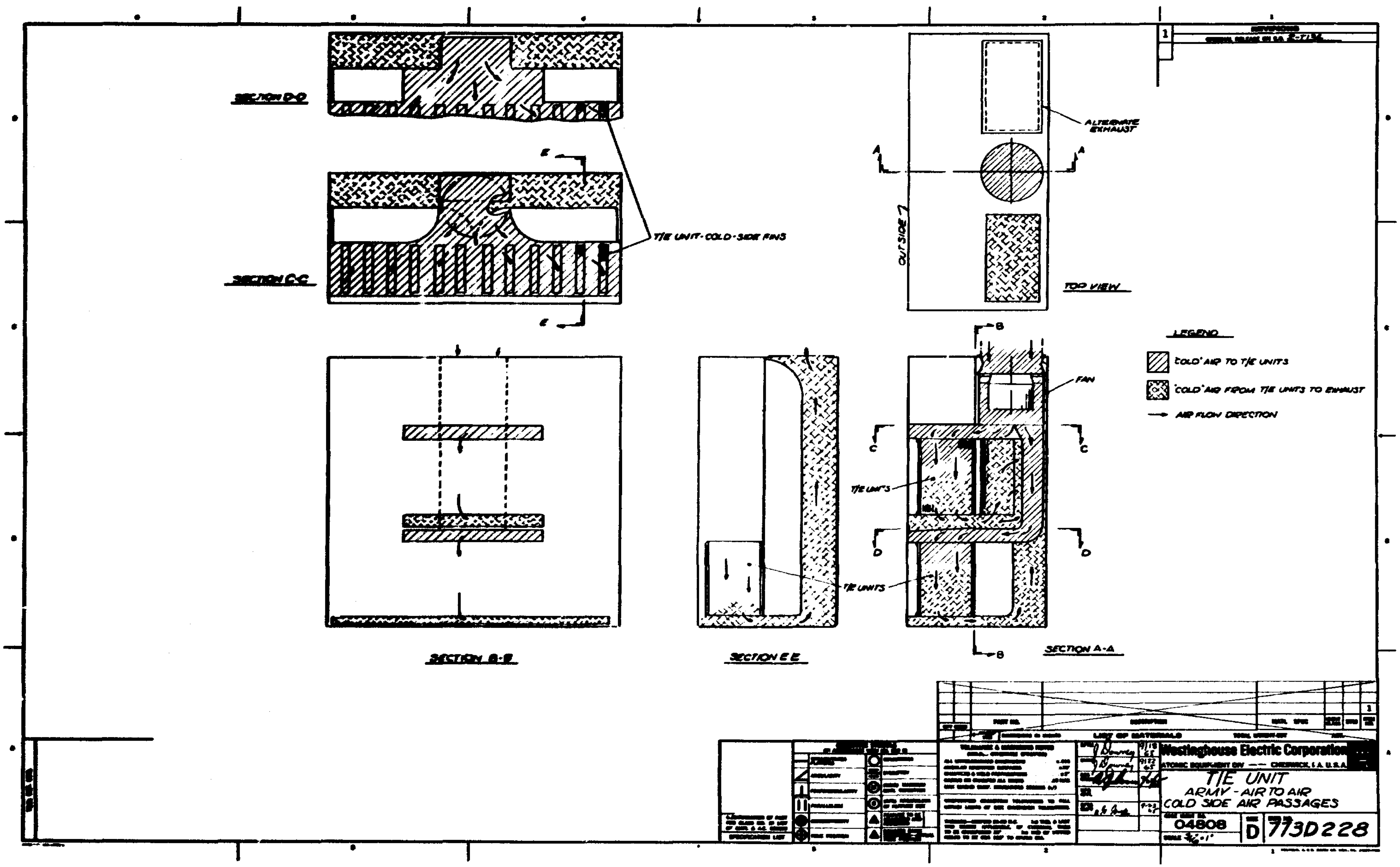
34808 J 616JK133

616JG33



CSM 1185

FIGURE 7-2



CSM 1191

FIGURE 7-3

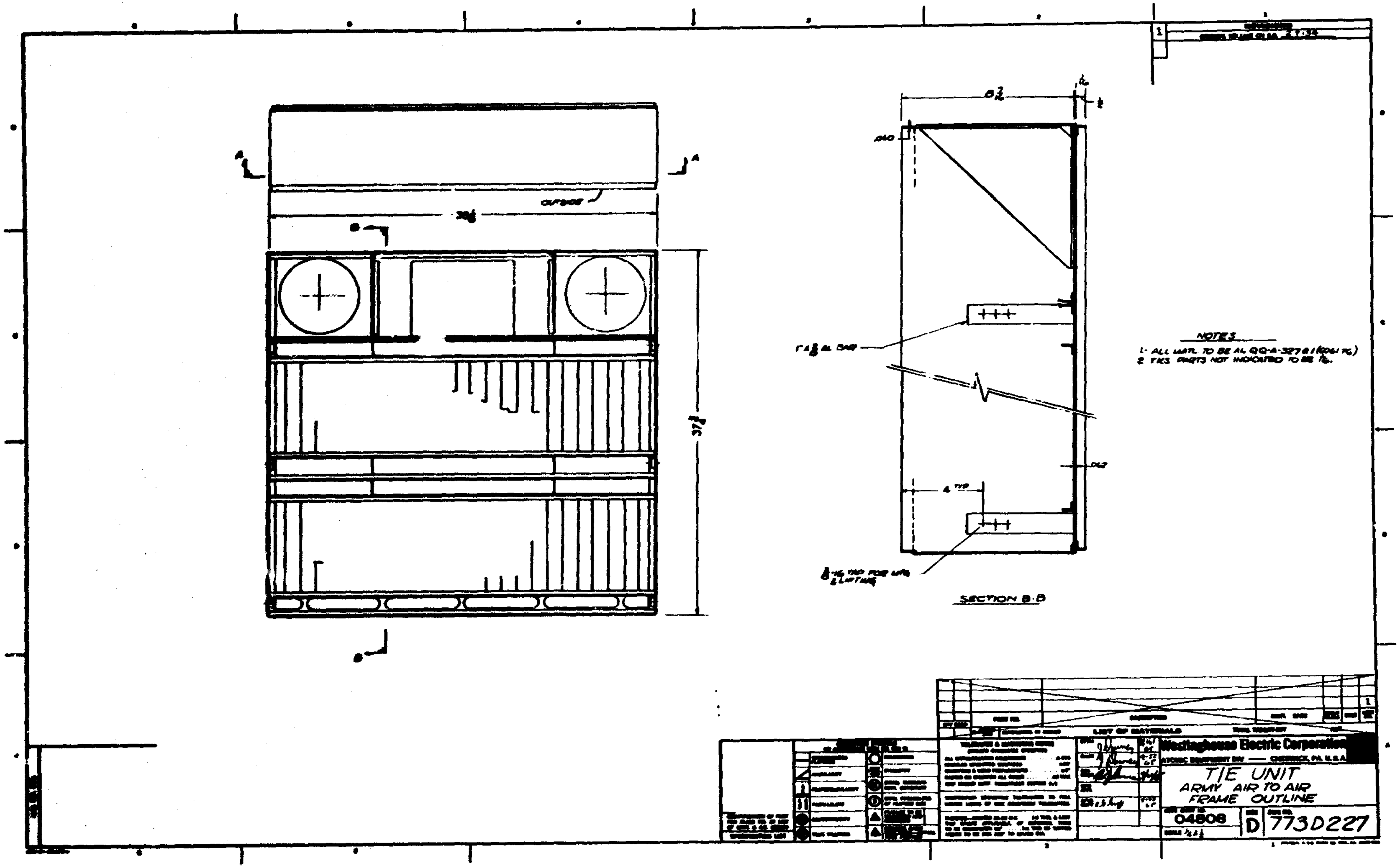
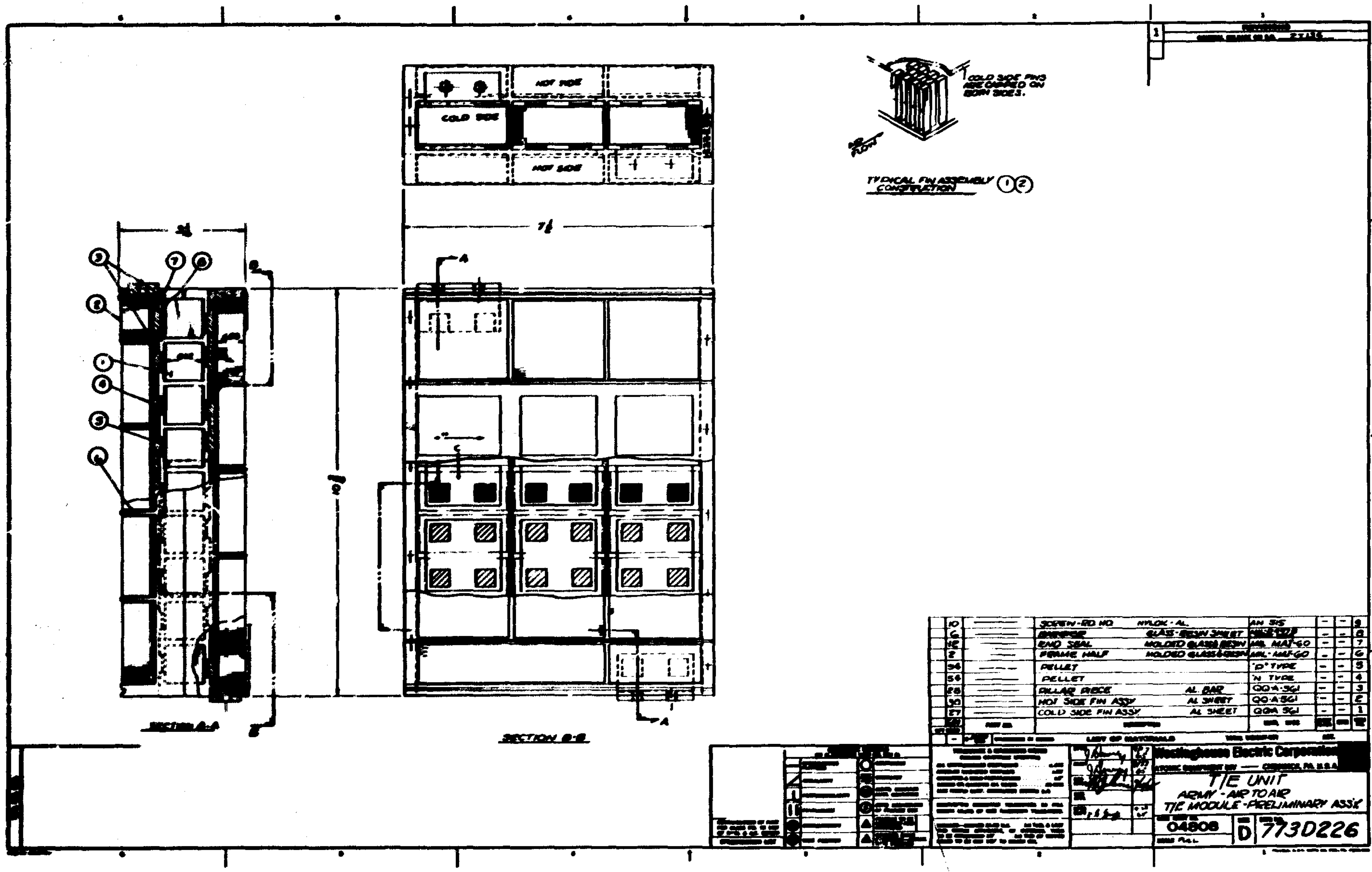


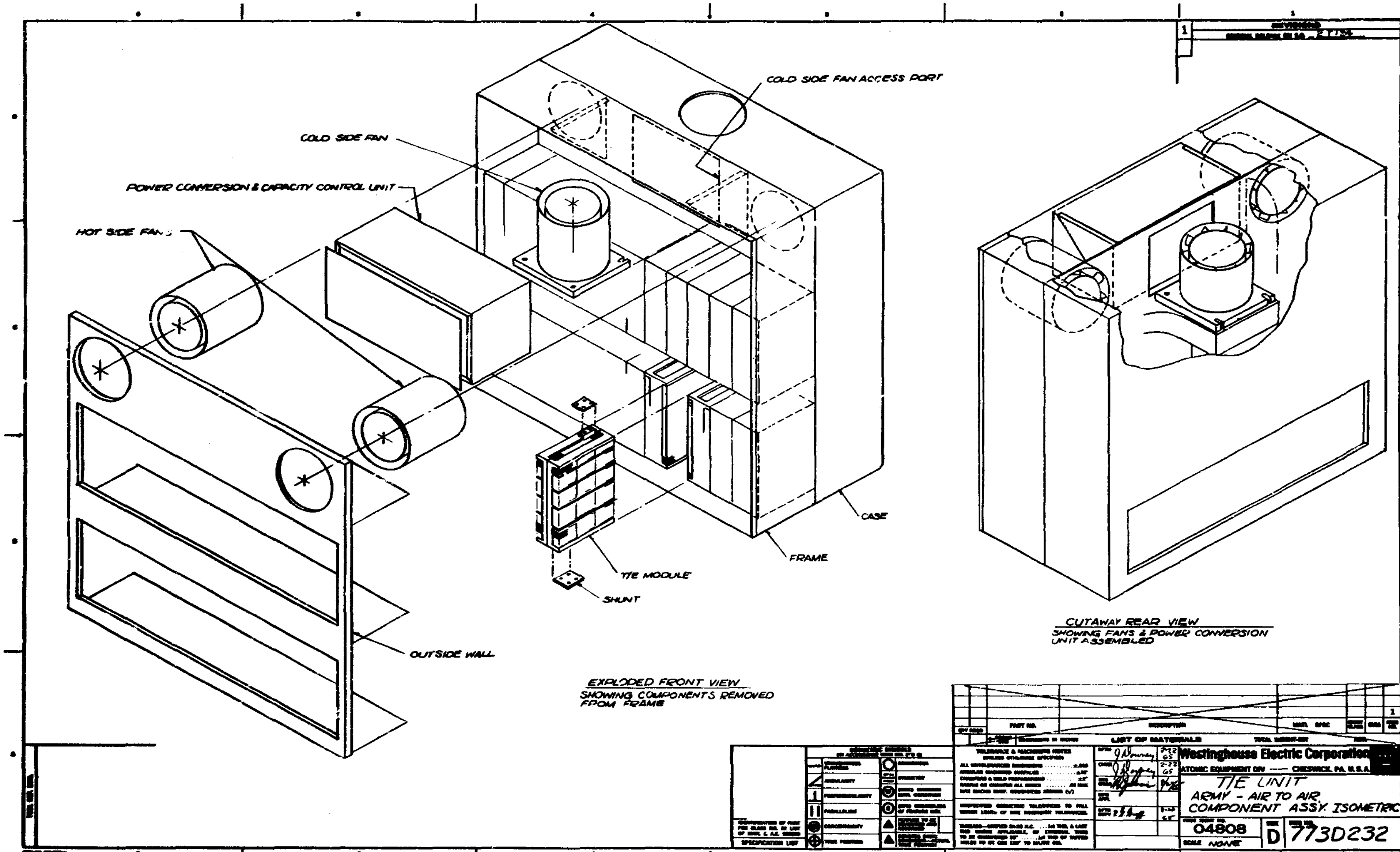
FIGURE 7-4

CSM 1186



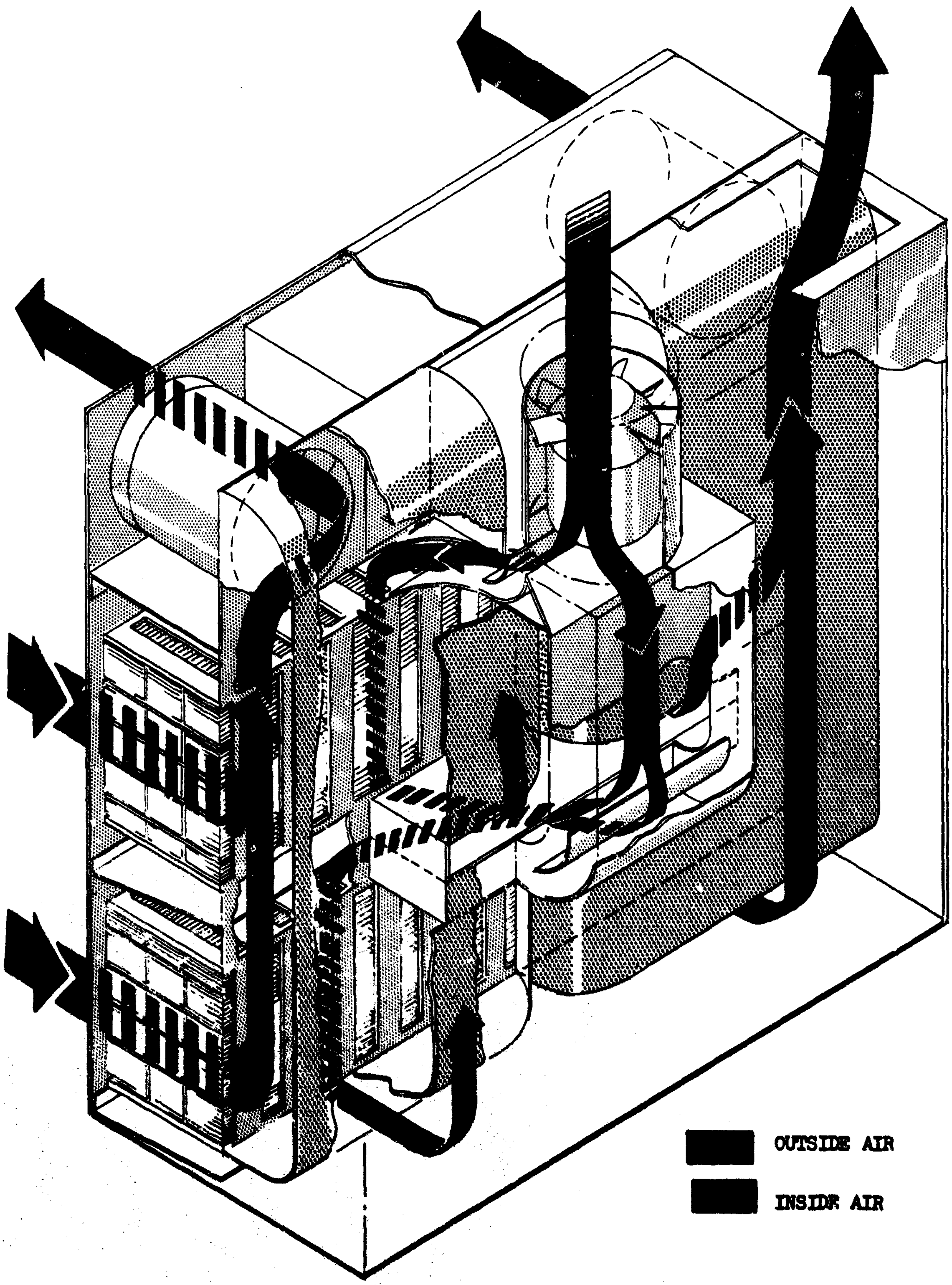
CSM 1168

FIGURE 7-5



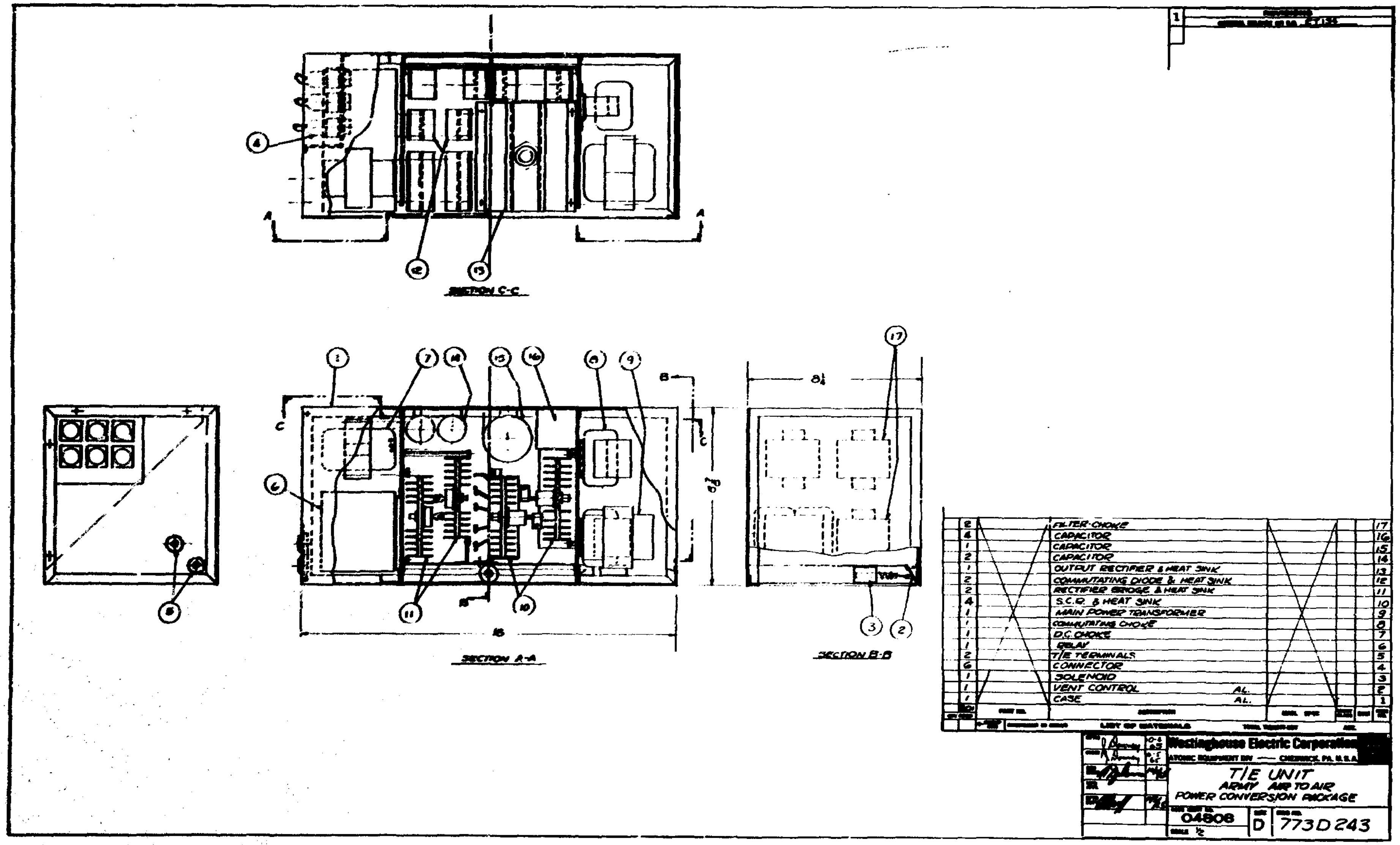
CSM 1187

FIGURE 7-6



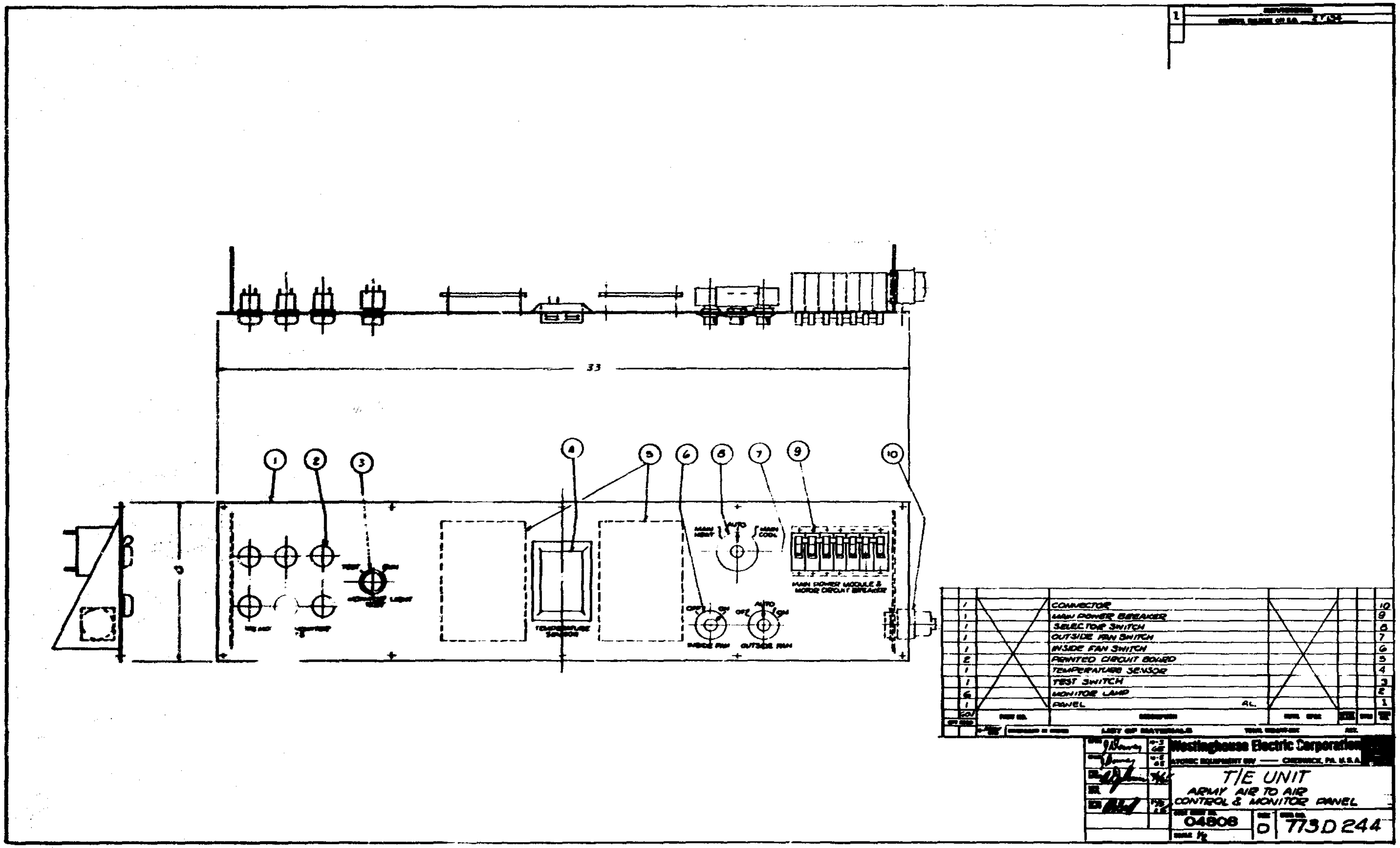
THERMOELECTRIC ENVIRONMENTAL CONTROL UNIT ASSY.
FIGURE 7-7

CSM 1163



CSM 1189

FIGURE 7-8



CSM 1190

FIGURE 7-9

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5. AUTHOR(S) *(Last name, first name, initial)*

R. E. Stillwagon
T. J. Dzakowic
W. P. Briley and G. T. Kartsovnaes

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Report on a design of a 24,000 BTU/hr. Thermoelectric environmental control unit.	U. S. AERDL Fort Belvoir, Virginia

13. ABSTRACT

An engineering design program was conducted for a Thermoelectric environmental control unit. The unit capacity is 24,000 BTU/hr. in the cooling mode when operating in 120° F ambient air. The unit is self contained, transportable and includes power conversion and automatic thermostatic control for both cooling and heating.

The final design is a result of design studies to optimize over-all system performance consistent with volume, weight, and cost considerations.

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